الجمهـــوريـــــة الجزائـــريـــــة الديمقـــراطيــــة الشـعبيــــة
République Algérienne Démocratique et Populaire
وزارة التـــعلـــيــــــم العــــــالـــــــي والبـــحــــــث العـلمـــــي
Ministère de l'Enseignement Supérieure et de la Recherche Scientifique

جامعة وهران 2 محمد بن أ حمد
Université d'Oran 2 Mohamed Ben Ahmed

معهد الصيانة و الأمن الصناعي
Institut de Maintenance et de Sécurité Industrielle

Département de Maintenance en Electromécanique

MÉMOIRE

Pour l'obtention du diplôme de Master

Filière : Electromécanique
Spécialité : Electromécanique Industrielle

Thème

Study of a Standing Wave Thermoacoustic Refrigerator using DeltaEC Software

Présenté et soutenu publiquement par :

Maouchi Toufik et Nemraoui Amine

Devant le jury composé de :

Nom et Prénom	Grade	Etablissement	Qualité
M. ADJELOUA Abdelaziz	MAA	Université d'Oran 2	**Président**
M. DARRAMDANE Mohamed Zouhir	MCB	Université d'Oran 2	**Encadreur**
M. ACHACHE Habib	MCA	Université d'Oran 2	**Examinateur**

Année 2020/2021

ACKNOWLEDGEMENT

First and foremost, we deem it a privilege to place on record the deep sense of gratitude to our supervisor ***Dr. DAR RAMDANE M.Z*** *for his patience, motivation, enthusiasm, and immense knowledge. His guidance helps us in all the time of research and writing of this work.*

A special thanks for the members of jury ***M. ADJELOUA AbdelAziz*** *and* ***M. ACHAHCHE Habib*** *for accepting the examination of our work.*

At last but not in least, we would like to thank everyone who helped and motivated us to finish this work.

Dedication

In the name of Allah, the most graciousmerciful and prophet Muhammad Sollallaahualaihi Wasallam, this work is dedicated to my family especially my father and my mother who have always been there in my life.

I would like to express my special thanks of gratitude to my supervisor Dr. ***DAR RAMDANE M.Z*** *who has guided and helped us in difficult periods. His motivation and help contributed tremendously to the successful completion of the project.*

I would also like to thank my friend ***Nemraoui Amine*** *and all my friends who helped me a lot in finishing this project Thanks again to all who helped me.*

Maouchi Toufik

Dedication

In the name of Allah, the most graciousmerciful and prophet Muhammad Sollallaahualaihi Wasallam, this work is dedicated to my family especially my father and my mother who have always been there in my life.

I would like to express my special thanks of gratitude to my supervisor ***Dr. DAR RAMDANE M.Z*** *who has guided and helped us in difficult periods. His motivation and help contributed tremendously to the successful completion of the project.*

I would also like to thank my friend ***Maouchi Toufik*** *and all my friends who helped me a lot in finishing this project*

Thanks again to all who helped me.

Nemraoui Amine

TABLE OF CONTENT

Introduction **1**

Chapter I Literature review

I.1 Introduction 3

I.2 Work carried out on TARs 4

I.2.1 Experimental works...4

I.2.2 Numerical simulation works 8

I.2.3 Theorical works:..9

I.3 Conclusion 10

Chapter II Theoretical study and operating principle

II.1 Theoretical study 12

II.1.1 Introduction 12

II.1.2 History...12

II.1.3 Thermoacoustic Heat Pumping 14

II.2 Thermodynamics Review..16

II.2.1 The First Law of Thermodynamics 16

II.2.2 Thermodynamic performance 16

II.2.3 Ideal Gasses 17

II.2.4 Length scales 17

II.3 Operating principle of a thermoacoustic refrigerator 18

II.3.1 Thermoacoustic refrigerator parts 19

II.4 Comparison of thermoacoustic refrigeration system with other refrigeration systems...22

II.4.1 Advantages, Inconvenient 24

II.5 Conclusion...25

Chapter III The Studied case

III.1 Introduction 27

III.2 Refrigeration systems...27

III.2.1 Vapor compression refrigeration cycle 27

III.2.2 Absorption refrigeration cycle...27

III.2.3 Brayton refrigeration cycle..28

III.2.4 Thermoacoustic refrigeration cycle 28

III.3 Deltaec description 29
III.4 Modeling in DELTAEC..........30
III.5 DELTAEC model..........30
III.6 Numerical simulation..........36
III.7 Conclusion..........37

Chapter IV Results and Discussion

IV.1 Introduction..........39
IV.2 Results and Discussion..........39
IV.3 Study of the optimal Stack position..........40
IV.4 Study of the mean pressure inside the resonator tube..........41
IV.5 Study of the acoustic pressure inside the resonator tube 43
IV.6 Effect of different gases on refrigeration performances 44
IV.7 Conclusion..........45

Conclusion 46

Abbreviation

TAR	Thermo Acoustic Refrigerator
VCRS	Vapor Compression Refrigeration System
CFC	Chlorofluorocarbon
LANL	Los Alamos National Laboratory
TADTAR	Thermoacoustically driven thermoacoustic refrigerator
AHX	Ambient heat exchanger
CHX	Cold Heat Exchanger
RPN	Reverse Polish Notation
COP	Coefficient of Performance
CFD	Computational Fluid Dynamics
DeltaEC	Design Environment for Low Amplitude Thermo Acoustic Energy Conversion.

Nomenclature:

δ_k	Thermal penetration depth
T_m	Mean temperature
P_m	Mean pressure
Q	The heat added to the system
W	The work done on the system
E	The total internal energy of the system
S	Entropy
P	Pressure
V	Volume
T	Temperature
R	Ideal gas constant
ρ	Density
γ	The ratio of specific heat
C_p	Heat capacity (constant pressure)
C_v	Heat capacity (constant volume)
λ	wavelength
c	speed of sound
f	frequency
L	length of resonator tube

Abstract

Development of refrigerators based on thermoacoustic technology is a novel solution to the present day need of cooling without causing environmental hazards. With added advantages of no moving parts and no circulating refrigerants, thermoacoustic refrigerator is a new and emerging technology capable of transporting heat from a low temperature source to a high-temperature source by utilizing the acoustic power input. The designing of thermoacoustic refrigerators is very challenging. This study illustrates the impact of significant factors on the performance of the thermoacoustic refrigerator in terms of the temperature difference generated across the stack ends. The results obtained show that the stack position, the acoustic and mean pressure have a big influence on the performances of the standing wave thermoacoustic refrigerator. The optimum position of stack is at 0.66m along the resonator tube and the best working fluid for the TAR is the Helium, because it required the minimum value of acoustic pressure of 0.285bar and a mean pressure of 10bar.

ملخص

يعد تطوير الثلاجات "الصوتية الحرارية" حلاً جديدًا في الوقت الحالي للتبريد دون التسبب في مخاطر بيئية. مع المزايا الإضافية لغياب الأجزاء المتحركة مقارنة بالمبردات المتداولة، تعد هذه الثلاجة تقنية جديدة وناشئة قادرة على نقل الحرارة من مصدر درجة حرارة منخفضة إلى مصدر درجة حرارة عالية من خلال استخدام مدخلات الطاقة الصوتية. يعد تصميم ثلاجات حرارية صوتية أمرًا صعبًا للغاية. توضح هذه الدراسة تأثير العوامل المهمة على أداء الثلاجة الحرارية الصوتية من حيث اختلاف درجات الحرارة الناتج عبر نهايات المبادل الحراري. تشير النتائج المتحصل عليها إلى أن موضع المكدس والضغط الصوتي والضغط المتوسط لهم تأثير كبير على أداء الثلاجة الحرارية الصوتية ذات الموجة الدائمة، الموضع المثالي للمكدس عند 0.66 متر على طول الأنبوب الرنان ويعد الهيليوم أفضل غاز لعمل الثلاجة "الحرارية الصوتية" الذي يحتاج إلى 0.285bar من الضغط الصوتي مع الضغط المتوسط بقيمة 10bar

Figures list

Chapter I

Figure I-1: (a) 3D printed stack with parallel geometry (b) Experimental layout [8]......................4
Figure I-2: Simplified model of thermoacoustic refrigeration [29]..7
Figure I-3: Solar energy driven TAR and its Frensnel lens by Adeff and Hofler [39] 8

Chapter II

Figure II-1: (a) Higgins' singing flame (b) The Rijke tube 12
Figure II-2: Hofler's standing-wave refrigerator. The hot end of the stack is thermally anchored at room temperature and the standing wave generates cooling at the cold end of the stack. 14
Figure II-3: Thermoacoustic heat pumping process 15
Figure II-4: thermoacoustic refrigerator device 18
Figure II-5: thermoacoustic refrigerator speaker...19
Figure II-6: Quarter wavelength Resonator 20
Figure II-7: Half wavelength Resonator 20
Figure II-8: (a) spiral, (b) parallel, (c) honeycomb, (d) pin array..21
Figure II-9: Heat Exchangers 22

Chapter III

Figure III-1: vapor compression refrigeration 27
Figure III-2: Absorption cycle 28
Figure III-3: Brayton refrigeration cycle. 28
Figure III-4: Thermo-acoustic refrigeration cycle 29
Figure III-5: DELTAEC User Interface 30
Figure III-6: schematic view of studied TAR using DeltaEC...30
Figure III-7: Studied TAR geometry...31
Figure III-8: DeltaEc model..32
Figure III-9: Segment 0 modeling parameters for the thermoacoustic refrigerator 32
Figure III-10: segment 1" SURFACE"..33
Figure III-11: segment 2 DUCT...33
Figure III-12: segment 3 heat exchanger HX 34
Figure III-13: segment 4 stack (STKLAB)..34
Figure III-14: Segment modeling the heat input to the cold side of the refrigerator from the outside 34
Figure III-15: the segment modeling cold duct...35
Figure III-16:second surface segment 35
Figure III-17: RPN segments..35
Figure III-18: segment 13 HARDEND 36

Chapter IV

Figure IV-1: acoustic pressure and velocity distribution in a standing wave thermoacoustic refrigerator..39
Figure IV-2: temperature distribution in standing wave thermoacoustic refrigerator 40

Figure IV-3: variation of temperature difference with respect of the stack position 40
Figure IV-4: variation of COP with respect of the stack position 41
Figure IV-5: Variation of temperature difference with respect to variation of the mean-pressure .42
Figure IV-6: variation of COP with respect of the mean-pressure variation.................................42
Figure IV-7: Variation of temperature difference with respect of variation of the acoustic pressure 43
Figure IV-8: Variation of coefficient of performance COP with respect of the acoustic pressure..43
Figure IV-9: Variation of acoustic pressure for different gases 44
Figure IV-10: Variation of frequency with gas type...45

List of tables

Table II-1: Comparison of thermoacoustic refrigeration system with other refrigeration systems ..23
Table II-2: Summary of advantages, inconvenient of thermoacoustic refrigerator 24

Introduction

Thermoacoustic refrigeration (TAR) is an application of thermoacoustic phenomenon that represents a solid-fluid interaction in the presence of acoustic waves. The development of TAR is important since it may replace current refrigeration technology, and it can be made with no moving parts. TAR are mechanically simpler compared to traditional vapor compression refrigerators (VCR) and do not require the use of harmful chemicals.

Our master memoir is divided into four chapters, the first chapter is devoted to the literature review where an update of works carried out on thermoacoustic refrigerators is presented. Many researchers perform studies on operating parameters and geometrical constraints in order to develop this technology. Most of the studies concerned with the stack which is an essential component of TAR, other works are related to the combination of working gas and stack material. Some researchers perform experiments, other deal with numerical simulation using DELTAEC and CFD tools.

The second chapter contains the theoretical study and the operating principle. In this chapter we start with thermoacoustic heat pumping process and thermodynamics Review. The operating principle of TAR that depends on its parts such as loudspeaker, resonator, stack, heat exchanger and working fluid is explained. A comparison of TAR with other refrigeration systems such as absorption vapor compression and thermo electric is done and at the end of this chapter a summary of the advantages and inconvenient of the TAR

Chapter three present the study of our case where we started with a general description of thermodynamic cycles, such as refrigeration vapor-compression, absorption, reversed Brayton cycles and thermoacoustic refrigeration. The detail of the studied geometry of the sanding wave TAR is presented and a design modeling of TAR is done using DELTAEC software with a description about our numerical simulation steps that has been applied to the model.

The fourth chapter is devoted to the discussion of the results obtained from the simulation where the influence of stack position, the impact of the mean pressure and acoustic pressure on the temperature difference and coefficient of performance is studied. The influence of different working fluid used in this thermoacoustic refrigerator is checked out.

Chapter I: Literature Review

I.1 Introduction

After successful flourishing thermoacoustic refrigerator (TAR) developed by Hofler in 1986 which had a capacity of 6-Watts, becomes an emerging cooling technology against the conventional Vapor Compression Refregiration System (VCRS) without any adversarial effect on the environment. However the TAR demanding the effective tuning between sound waves and resonator tube frequency and efforts are made in a diverse course to overwhelmed these issues. First commercial TAR developed [1] with system operating pressure of 10 atm and cooling capacity of 119 Watts for ice cream sales.

Investigators are trying to focus on operating parameters and geometrical constraints and most studies concerned with the stack which essential component of TAR due to which temperature gradient exist. It plays a vital role in producing a thermoacoustic effect, the other work related to a combination of working gas and stack material [2] [3] [4]. Some researchers perform numerical simulation using DELTAEC and CFD, such ANSYS tool. Now a day's researchers try to eliminate loudspeaker that is a source of acoustic waves with solar energy, waste heat from industries or power plant. This low-grade energy used to generate acoustic power the resulting TAR known as heat powered TAR [5]. The absence of moving component and use of non-hazardous inert gasses as their working fluid makes TAR compact and reliable. TAR finds an application in the area of food preservation, transportation of perishable products; space operation [6] small capacity electronic equipment. The Thermoacoustic refrigeration can be a promising cooling technology in remote areas where electricity is a critical issue. A passenger vehicle can be air conditioned by waste heat from the exhaust of engine [7] considering a TAR.

I.2 Work carried out on TARs

I.2.1 Experimental works

A standing wave thermoacoustic stack fabricated using a 3D printer has been optimized by **Nor Atiqah Zolpakara** et al.to determine the optimum design parameters; stack length, center position and plate spacing. The results show that the 3D-printed stack has potential towards improvement of the temperature performance of the thermoacoustic refrigeration system, although a more refined fabrication technology is still in need. They realized that 3D-printing of the stack minimizes the error, eliminates inconsistencies, and reduces the time for the production of the final product. [8]

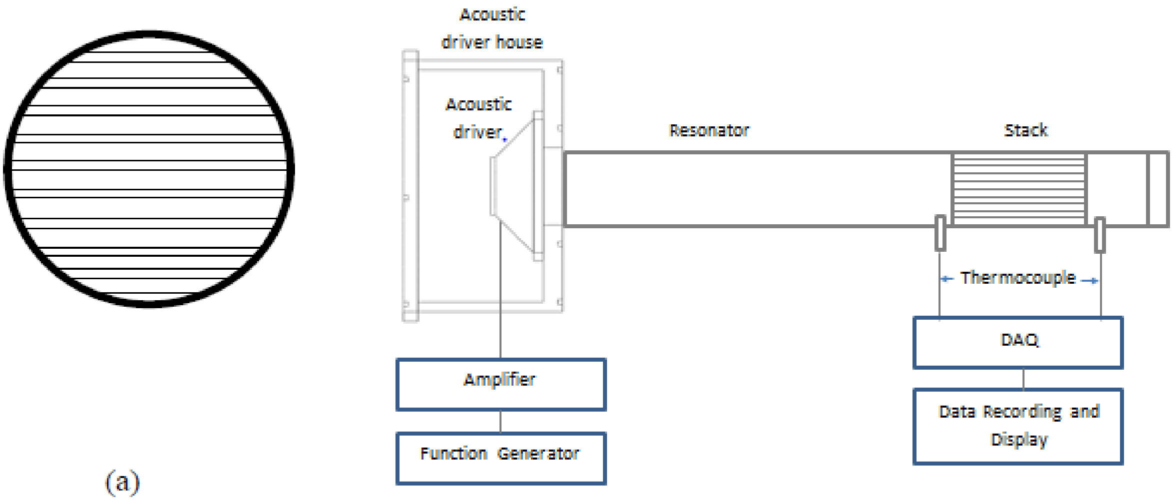


Figure I-1: *(a) 3D printed stack with parallel geometry (b) Experimental layout* [8]

Jonathan Newman et al. [9] show that thermo-acoustic device is possible and able to cool airand demonstrate that it has the ability to create and maintain a large temperature gradient, more than 20 degrees Centigrade, which would be useful as a heat pump. [10]

An experiment had been carrying out by **L.K Tarbitu [11]** where a honeycomb structure as stack geometry with air as working gas is considered with different position of square shape stack. They considered a quarter wavelength resonator tube with six different positions of the stack. The analytical results were closer to the experimental and a maximum COP observed when stack placed near the pressure antinode. [12]

MEH Tijani et.al [13] built a thermoacoustic cooler comprises of hot and cold heat exchanger with parallel stack geometry and helium as working gas in resonator tube An experiment performed with two different structure of stack (parallel and spiral), it reveals that parallel geometry performs better than spiral geometry. The lowest temperature of -67 °C reached with COP of 11 % that was about 20 % of Carnot COP. [12]

G.Allesina [14] designed and built a standing waves refrigerator with more attention on woofer box containing loudspeaker, insulation around the stack to reduce the losses.. The disturbance of sound waves was common when resonator tube becomes narrow. [12]

E.C.Nsofor et al. [15] examine the effect of operating pressure and frequency on the performance of TAR, to reduce the axial heat conduction within stack a whole resonator made by aluminum tube covered with the plastic tube. The authors had discussed the effect of a temperature gradient within the stack, with an increase in temperature gradient the cooling load increases. The main intention of this study is to optimize the operating frequency and pressure which result in higher cooling load and improved system performance. [12]

N.Yassen [16] modified a design strategy and built standing waves TAR powered by solar energy. The system comprises of PhotoVoltaic cell, stack, sound generator and two heat exchangers. [12]

A study of the Thermoacoustic system was conducted by **Pranav Mahamuni et** al [18]. The model was tested for various frequencies from 250 Hz to 500 Hz in steps of 50 Hz. The results obtained indicate that the temperature at the hot end increases with the frequency and then remains stable.

The impact of the resonance tube on the performance of the thermoacoustic stack was studied by **Channarong Wantha [19]**. The key component of a standing wave TAR is the resonance tube. The performance of the system increases with appropriate resonator length. Results obtained indicate that the optimal operating frequency and that obtained from the design based on the equations of half wavelength differs from each other.

The stack is one of the most important parts of a thermoacoustic system that affects its performance. **A. C. Alcock** investigated the performance of the ceramic substrates used as a stack material in standing wave thermoacoustic refrigeration **[22]**. The geometric configuration of the stack influences the performance of the system which was studied. The porosity of the stack, its length and the position of the stack in the resonator were varied. Another important consideration is the material choice of the stack and the method of manufacturing. Results obtained indicate that the resonant frequencies were different for each position of the stack. It was observed that the difference in temperature becomes maximum, when the stack is positioned closer to the pressure antinode. The length of the stack had very insignificant effect on the performance of the device. [17]

Stack is the medium of heat transfer and is the heart of the thermoacoustic system. An overview of the design of the stack is presented by **Bhansali. P. S [23]**. Thermal properties such as thermal conductivity and the specific heat are important for the design of stack. Optimal spacing of the stack plates based on the thermal penetration depth and the viscous penetration depth are discussed. Stack made of vitreous carbon in various geometries like parallel plate, porous type, spiral type and parallel plate type. The important characteristics of the stack are high heat capacity and low thermal conductivity. Stack geometry and spacing play a major role in the refrigeration. Very less spacing gives rise to high viscous losses whereas more spacing leads to less volume of gas to involve in thermal interaction. Optimal stack spacing of three times the thermal penetration depth is suggested. [17]

A small scale thermoacoustic refrigeration system was fabricated and studied experimentally by **B. Ananda Rao [24]**. Through his experiment, he found that the variation in the temperature difference across the stack was under 0.5°C when there was no stack placed inside the resonator. When the stack was used, the difference in temperature of 15°C was obtained after 300 seconds. The maximum difference in temperature was found to occur when the stack is placed at the pressure antinodes. The insulation around the resonator was found to have very less effect on the obtained refrigeration. [17]

Ikhsan Setiawan conducted an experimental study on the thermoacoustic cooling system using two stacks in a straight resonator tube **[25]**. The resonator was made of PVC of length 112cm using air at atmospheric pressure at the frequency of 152 Hz. Stacks are made of parallel plate type with spacing about four times the thermal penetration depth. It was observed that the maximum temperature difference occurs when the stacks are placed at the ends of the tube near the pressure antinodes at a frequency of 147 Hz. Higher temperature occurs at higher input power. [17]

The effectiveness of the thermoacoustic refrigeration system was examined by **Kaushik S. Panara [26] et B. Ananda Rao[24]. Kaushik S. Panara** brings out that the performance of the refrigerator depends on the working gas, the shape of the resonator, pressure inside the resonator tube, stack material, length and its position. Thermoacoustic refrigeration system offers the combined benefits of providing heating and cooling simultaneously. Results obtained indicate that a variation in the temperature of only 0.5°C was observed inside the resonator tube without the stack. Presence of stack creates a difference in temperature. The position of the stack is of essential importance in order to get maximum temperature difference and it occurs at the pressure antinodes. The input power as well as the effectiveness of the acoustic driver plays a major role in creating the maximum temperature gradient. The insulation around the resonator has very little effect on the temperature difference. [17]

The methods to improve the performance and the temperature difference were detailed by **Jithin George [27]**. Increasing the length of the resonator causes an increase in the temperature at the hot end of the stack. The increased temperature difference can be achieved by the use of heat exchangers at the ends of the stack. A stack made of low conductivity material would enable to reduce the heat diffusion across the stack. Optimum spacing of the plates in the stack is also essential for the improved performance. [17]

Ramesh Nayak. conducted an experiment to evaluate the performance of thermoacoustic refrigeration system under various operating conditions and stack geometries **[28]**. His results indicated that higher temperature difference was obtained using parallel plate stack. The temperature difference increases as the heating load increases. Small Thermoacoustic refrigeration system was constructed with inexpensive and readily available parts by SreeneshValiyandi [41]. He suggested that in order to obtain higher cooling effect higher heat carrying capacity materials could be used and inert gases could be used as working fluid. [17]

A simplified model of a TAR was given by Hofler, shown in Figure I-2. The refrigerator system mainly includes a gas-filled resonator, a driving loudspeaker, and a stack of plates with one end thermally anchored at room temperature. The “stack of plates” was a long strip of plastic (Kapton) sheet, spirally wound around a plastic rod with a spacing between the layers of plastic sheet (roughly 4 times of thermal penetration depth). Each of the two heat exchangers was made of rectangular copper strips reaching across the stack ends. The “Hofler resonator”, comprising a largediameter section, a small-diameter section and a sphere in series, was essentially a quarter wavelength resonator. [40]

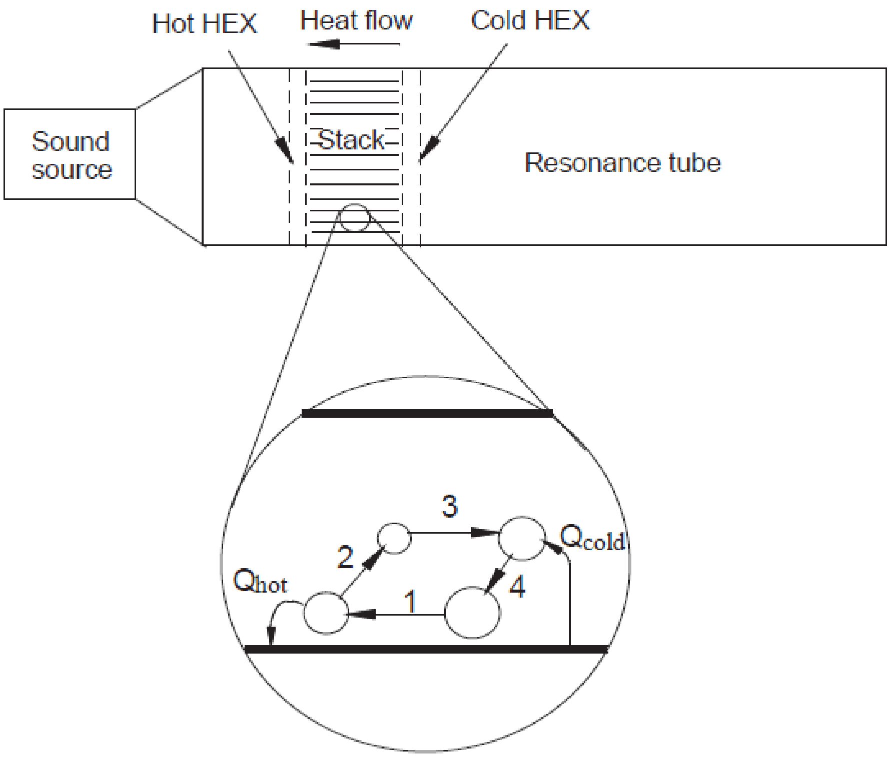


Figure I-2: Simplified model of thermoacoustic refrigeration [29]

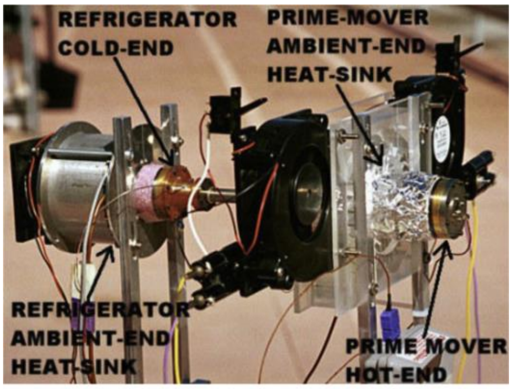
The influence of wave patterns and the frequency on the thermoacoustic cooling effect was investigated by **Yousif A. Abakr [31].** Simple TAR system was designed and tested for the effects of wave patterns and frequency on the cooling. It was observed that the square wave pattern yielded better results when compared with other wave patterns. [17]

Maxime Perier- Muzet designed and analysed the dynamic behaviour of a cold storage system combined with a thermoacoustic system that is solar powered **[35]**. A thermoacoustic engine produces acoustic work by the utilization of the waste heat in a heat powered TAR. This is coupled to a thermoacoustic cooler that enables the conversion of acoustic energy into cooling effect. The study has demonstrated the usage of solar energy as the source of thermal energy for a low power heat driven TAR. [17]

M. Nouh constructed a piezo driven TAR with dynamic magnifiers **[36]**. The magnified refrigerator demonstrated the effectiveness of piezoelectric actuation.

Although the current applications of thermoacoustic refrigeration system are limited, with further studies and investigations, this method of refrigeration can be used in many domains. Some of the areas where the system can be potentially applied are in the liquefaction of natural gas [38], in obtaining heating and cooling simultaneously [30], utilization of waste heat in industries [37]. **J. A. Mumith** designed a thermoacoustic heat engine for the low temperature waste heat recovery in food manufacturing [37], for cooling applications in space (STAR Space Thermo Acoustic Refrigerator), cooling of electronic components in ships and in cooling of ice creams. [17]

In 2000, **Adeff et Hofler [39]** built and tested a solar thermal powered TADTAR (thermoacoustically driven TAR) of a small temperature span shown in figure I-3. The higher-intensity sound waves then drove the TAR to achieve a substantially improved cooling power and the temperature span had also been enlarged from 18 to 30 °C. [40]



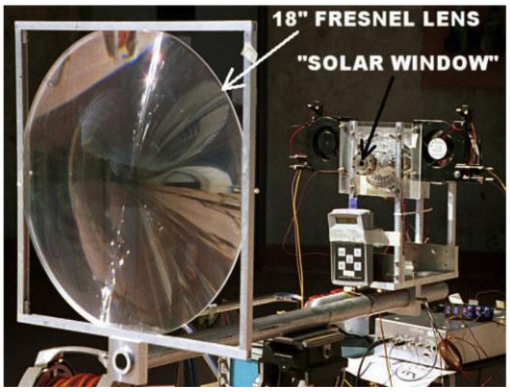


Figure I-3: *Solar energy driven TAR and its Frensnel lens by Adeff and Hofler* [39]

I.2.2 Numerical simulation works

Ali Namdar et al [20] try to simulate input pressure using open foam package in CFD, here the maximum temperature and velocity profile has studied, and the main conclusion of this study is optimum left side heat exchanger position for enhanced performance of TAR. It was observed that with lower oscillation pressure the heat has not rejected to atmosphere via hot heat exchanger while at higher oscillation pressure the gas parcel not absorb the heat from the left side of the heat exchanger with a sharp rise in nonlinear effects. [12]

A standing wave TAR was designed by using a numerical approximation with the use of a modelling code called as Design Environment for Low Amplitude Thermo Acoustic Energy Conversion (DeltaEC). The configuration of the stack such as diameter, length, porosity and position of the stack has significant effect on the performance of the system. The main objective of the work is to identify and select the best geometric configuration of the stack. By simulating using DeltaEC, the performance in terms of COP was evaluated by varying geometric configurations of stack and the best condition was selected. [17]

An algorithm based on simplified linear TA model was developed by **Hadi Babaei** that has the ability to design thermoacoustically driven refrigerators **[32]**. This is a new feature based on the energy balance to design a thermoacoustic engine and acoustically driven refrigerators. It has been found that the results of the algorithm are in agreement with the results obtained from the computer code DeltaEC. [17]

The CFD analysis of thermoacoustic cooling was done by **Florian Zink [33]**. In thermoacoustic energy conversion, the application of numerical analysis techniques, particularly, Computational Fluid Dynamics (CFD) Analysis has gained importance. The efforts made previously focused on single thermoacoustic couples that were subjected to thermoacoustic effect using an oscillatory boundary condition. It is computationally expensive to conduct CFD analysis of the entire thermoacoustic system. The prescribed work examines the simulation of an entire thermoacoustic engine that includes thermoacoustic refrigeration. The cooling of the working gas in the stack is demonstrated by the interaction of thermally generated sound waves. Temperature reduction below the ambient temperature was simulated by the new model and it was found to be consistent with the similar physical model. Location of the stack near the pressure node is of prime importance as it should yield better results. [17]

I.2.3 Theorical works:

The acoustic coupling between the loud speaker and the resonator in a standing wave Thermoacoustic drive was explained by **David Marx [21]** usinglinear acoustic equations and linear model of the loud speaker. For low values of excitation voltage corresponding to the low acoustic pressure, the comparisons with the measurements are good. However, the measured drive ratios are found to be lower than predicted when there is an increase in the voltage. [17]

The parameter estimation for the characterization of thermoacoustic stacks and regenerators was done by **Matthieu Guedra [34]**. The study deals with the in-situ characterization of open cell porous stack. An inverse method is used to estimate the geometric and the thermal properties of the stack surrounded by the heat exchangers and connected to the thermal buffer tube to form the thermoacoustic core. The experimental data obtained from different stacks under varying heating conditions are used to fit the theoretical forward model by adjusting the geometric parameters of sample and heat exchanger coefficients. It was found that carbon foam allows getting higher temperature gradients. [17]

Finally, thermoacoustic refrigeration systems offer many advantages over the conventional refrigeration systems. Thermoacoustic system has the potential to develop renewable energy systems by the utilization of waste heat or solar energy as stated by HadiBabaei [32].

I.3 Conclusion

This chapter presents the various important studies that have been made on the thermoacoustic refrigeration system which can serve in the design of the TAR. These studies focused on improvement and optimization of the performances of the TAR. Most of the studies concerned with the stack which is an essential component of TAR due to the generation of temperature gradient inside it. Other work related to a combination of working gas and stack material. Some researchers perform numerical simulation using DELTAEC and CFD, tools. Recently different algorithms are also used to augment the different parameters of TAR.

Chapter II:

Theoretical study and operating principle

II.1 Theoretical study

II.1.1 Introduction

Thermoacoustics is a subject which focuses on the conversion of thermal energy into acoustic energy and vice versa in the present of solid walls and oscillating fluids inside a resonance tube. In 1980, Nikolaus Rott [42] first introduced the term “thermoacoustics” in a review of his previous work on a theory for this phenomenon. The theory, known as linear thermoacoustic theory, became the solid basis of nowadays thermoacoustic investigations and applications. In recent decades, investigations on the fundamental nature of the problems encountered in various thermoacoustic devices and explorations on industrial and household applications are widely carried out in many research groups world-wide.

Many thermoacoustic devices were built and utilized. The performances and efficiencies are much enhanced. Thermoacoustic devices have the advantages over the conventional heat pumps and engines, that they have no mechanical moving parts, which brings high reliability and virtually maintenance-free to the customers.

Moreover, they are environment friendly by using chemically inert working gases. It is a charming technology for today’s world, which is suffering all sorts of environmental problems: global warming, ozone depletion and others. [43]

II.1.2 History

Thermoacoustics has a long history that dates back more than two centuries.

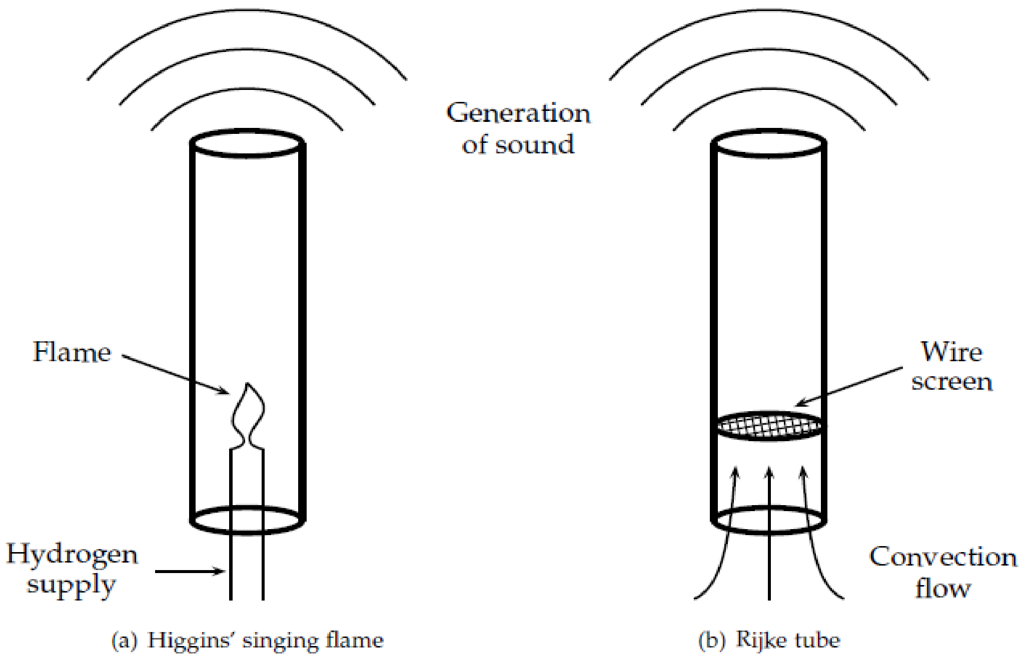


Figure II-1: *(a) Higgins' singing flame (b) The Rijke tube*

The first records of heat-driven oscillations are the observations of Higgins [44, 45] in 1777, who experimented with an open glass tube in which acoustic oscillations were excited by suitable placement of a hydrogen flame, the so-called “singing flame”.

A similar, but more famous experiment was performed by Rijke [46] who in his efforts to design a new musical instrument from an organ pipe, constructed the so-called “Rijke tube”. As depicted in figure II-1, he replaced Higgins’ hydrogen flame by a heated wire screen and found that when the screen was positioned in the lower half of the open tube spontaneous oscillations would occur, which were strongest when the screen was located at one fourth of the pipe.

The first qualitative explanation for heat-driven oscillations was given in 1887 by Lord Rayleigh. In his classical work “The Theory of Sound” [47], he explains the production of thermoacoustic oscillations as an interplay between heat fluxes and density variations:

”If heat be given to the air at the moment of greatest condensation (compression) or taken from it at the moment of greatest rarefaction (expansion), the vibration is encouraged”.

Rayleigh’s qualitative understanding turned out to be correct, but a quantitatively accurate theoretical description of these phenomena was not achieved until much later.

In 1964 Gifford and Longsworth [48] invented the pulse-tube refrigerator, by which they managed to cool down to a temperature of 150 K. Nowadays pulse-tube refrigeration is one of the most favored technologies for cryocooling.

In the eighties a very intensive and successful research program was started at the Los Alamos National Laboratory by Wheatley, Swift, and coworkers [49, 50, 51]. Using Rott’s theory of thermoacoustic phenomena they started to design and build practical thermoacoustic devices. Important was Hofler’s invention of a standing-wave thermoacoustic refrigerator [52, 53], which proved that Rott’s theoretical analysis was correct. Hofler’s refrigerator, shown in figure II-2, used a loudspeaker to drive a closed resonator tube with a stack of plates positioned near the speaker. At the other end of the tube a resonator sphere was attached to simulate an open ending, so that effectively one can speak of a quarter-wave-length resonator. Inside the refrigerator a standing wave is maintained by the speaker, generating a temperature difference across the stack such that heat is absorbed at the low temperature or waste heat is released at the high temperature.

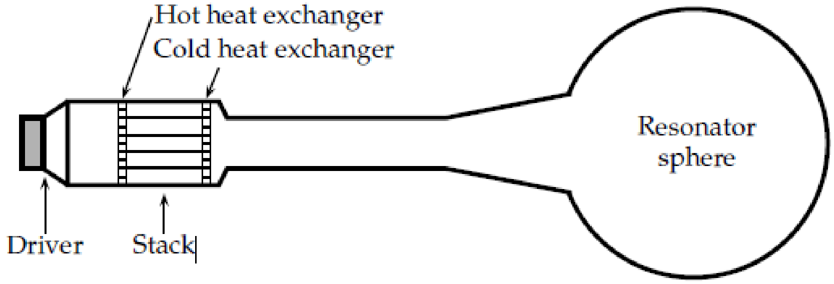


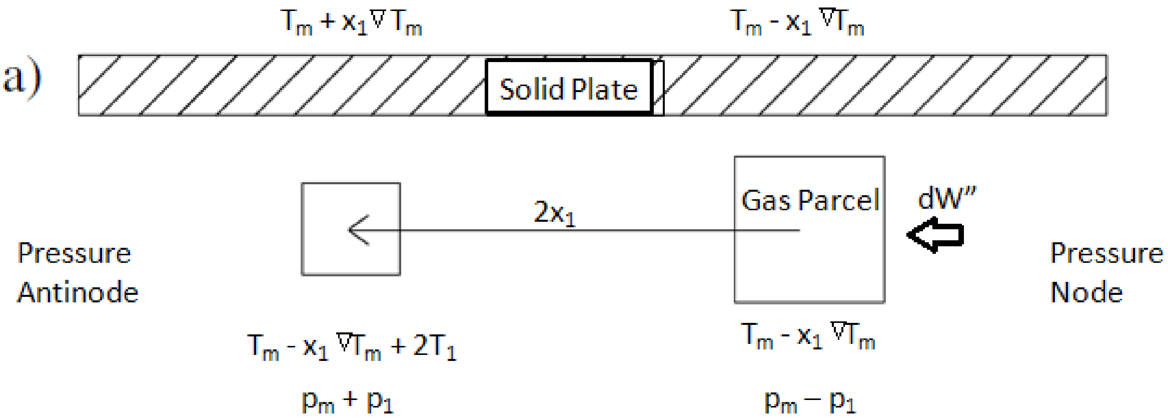
Figure II-2: *Hofler's standing-wave refrigerator. The hot end of the stack is thermally anchored at room temperature and the standing wave generates cooling at the cold end of the stack.*

A whole new branch of thermoacoustic devices started in 1979 with Ceperley's realization [54, 55] that thermoacoustic devices based on the Stirling cycle [56] with ideal heat transfer, could reach much higher efficiencies than devices based on standing wave modes of operation. His idea was to design machines that allow a traveling wave to pass through a dense porous medium (the regenerator) using a toroidal geometry. [57]

II.1.3 Thermoacoustic Heat Pumping

The formation of temperature gradient due to acoustic oscillations along the length of a plate can be understood from figure II-3 [58, 59]. Consider a solid plate placed in an acoustic field with direction of particle oscillation along its length. Suppose the pressure antinode (region of maximum pressure variation) is near the left end of the plate and the pressure node (region of zero pressure variation) is near the right end of the plate. the plate as well as the gas is at the mean temperature 'Tm' and the gas is at the mean pressure of 'p_m'. A typical gas parcel oscillates over a distance $2x_1$ about its mean position. Its pressure varies between $p_m - p_1$ and $p_m + p_1$. The activity of a typical gas parcel is shown below in Figure II-3 (a-d).

a)



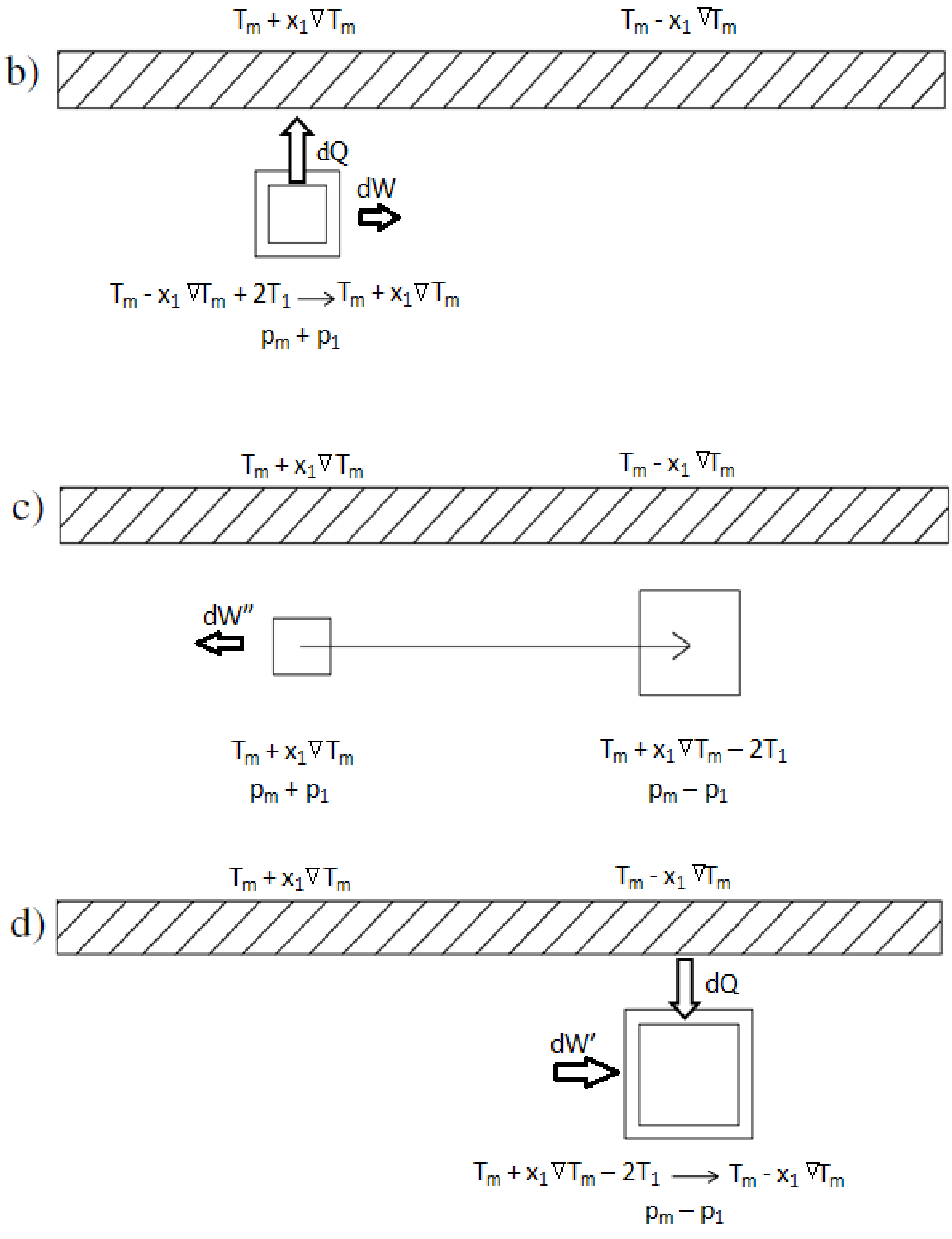


Figure II-3: *Thermoacoustic heat pumping process*

Referring to figure II-3(a), the gas parcel at the right end absorbs acoustic power and moves by a distance ‘$2x_1$’ to the left. During this displacement it gets compressed and its temperature rises. This parcel then loses heat to the plate till its temperature equals that of the plate (Figure II-3(b)). As a result, left end of the plate becomes a little warmer. The parcel then moves again to right end where its pressure as well temperature falls (Figure II-3 (c)). The cold parcel warms up by picking heat from right end of the plate, making the right end of the plate colder (Figure II-3(d)). Thus, in one cycle the gas transports 'dQ' amount of heat over a temperature difference of '$2x_1\nabla Tm$', absorbing 'dW` - dW' amount of acoustic power. Eventually, the left end of stack heats up and the right end cools down.

II.2 Thermodynamics Review

The study of heat engines and refrigerators constitutes a major portion of the field of thermodynamics. Therefore, it should come as no surprise that thermodynamics is at the heart of thermoacoustic theory, since the applications of the thermoacoustic effect are heat engines and refrigerators. [70]

II.2.1 The First Law of Thermodynamics

The first law of thermodynamics states that energy cannot be created or destroyed. Energy is simply transformed from one type to another. The total energy of a system is changed only by energy in the form of heat being added to the system or through work being done on the system. Note that heat was shown to be a form of energy by Joule in the mid nineteenth century [70, 71]

$$\boldsymbol{\delta Q + \delta W = dE} \qquad \text{II-1}$$

In equation (II-1), Q is the heat added to the system, W is the work done on the system, and E is the total internal energy of the system.

When dealing with thermoacoustic applications, most of the heat transfer will occur through conduction. The Second Law of Thermodynamics

The first law of thermodynamics is not violated in either case, and so the second law of thermodynamics is necessary to describe which energy flows and transformations will take place and which will not.

Every system has a scalar quantity called *entropy*. This quantity is related to how useable the energy within that system is. The entropy of a system is decreased if heat is removed from the system to the outside.

Energy transformations can be broken down into two categories. There are reversible energy transformations and irreversible energy transformations. [70]

$$\boldsymbol{dS = \frac{dQ}{T} + (dS)_{gen}} \qquad \text{II-2}$$

Where: $(dS)_{gen} \geq 0$

II.2.2 Thermodynamic performance

The performance of a refrigerator is measured by the so-called coefficient of performance COP.

When analyzing a refrigerator, we are interested in maximizing the cooling power $\dot{Q}c$ extracted at temperature T_C, while at the same time minimizing the net required acoustic power $\dot{W}$.

$$\boldsymbol{COP_{ref} = \frac{\dot{Q}_c}{\dot{W}}} \qquad \text{II-3}$$

The second law of thermodynamics limits the interchange of heat and work.

$$\boldsymbol{COP_{ref} = \frac{\dot{Q}_c}{\dot{Q}_H - \dot{Q}_c} \leq \frac{T_c}{T_H - T_c} = COPC_{ref},} \qquad \text{II-4}$$

The quantity COPC is called the Carnot coefficient of performance, and gives the maximal performance for all refrigerators. Using COPC we can introduce a relative coefficient of performance COPR as [69]

$$\boldsymbol{COPR = \frac{COP}{COPC}} \qquad \text{II-5}$$

II.2.3 Ideal Gasses

These working fluids are generally able to be considered ideal gasses. Ideal gasses follow a few important and well-known relations which are shown below

$$\boldsymbol{p = \rho RT} \qquad \text{II-6}$$

$$\boldsymbol{C_p = \frac{R\gamma}{\gamma - 1}} \qquad \text{II-7}$$

$$\boldsymbol{C_v = \frac{R}{\gamma - 1}} \qquad \text{II-8}$$

Equation (II-7) is known as the ideal gas law, equation (II-8) is the constant pressure specific heat, and equation (II-9) is known as the constant volume specific heat. Also, γ is defined as the ratio of specific heats and it can be shown by combining equations (II-7) and (II-8) that it is equal to the ratio of the constant pressure specific heat to the constant volume specific heat. [69]

II.2.4 Length scales

The length scales of a thermoacoustic device play an important role in its performance. The important length scales are the wavelength λ, frequency and speed of sound: [70]

$$\boldsymbol{\lambda = \frac{c}{f}} \qquad \text{II-9}$$

$$f = \frac{c}{2L} \quad \text{II-10}$$

Where: λ = 2L

$$c = \sqrt{\gamma r T} \quad \text{II-11}$$

Where: $r = \frac{R}{M}$

II.3 Operating principle of a thermoacoustic refrigerator

Thermo acoustic refrigerator is a device that uses energy of sound waves or acoustic energy to pump heat from low temperature reservoir to a high temperature reservoir. The source of acoustic energy is called the driver which can be a loudspeaker. The driver emits sound waves in a long hollow tube filled with gas at high pressure. This long hollow tube is called as resonance tube or simply resonator. The frequency of the driver and the length of the resonator are chosen so as to get a standing sound wave in the resonator. A solid porous material like a stack of parallel plates is kept in the path of sound waves in the resonator. Due to thermo acoustic effect heat starts to flow from one end of stack to the other. One end starts to heat up while other starts to cool down. By removing heat by means of a heat exchanger from the hot side of stack, the cold end of stack can be made to cool down to lower and lower temperatures. A refrigeration load can then be applied at the cold end by means of a heat exchanger.

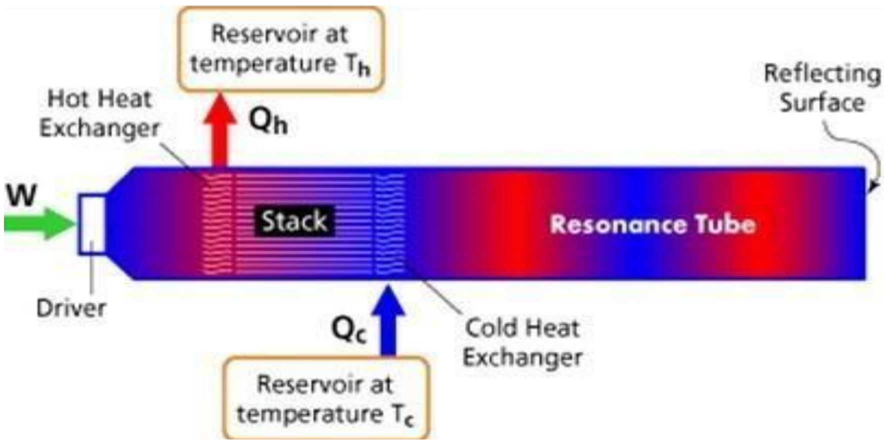


Figure II-4: *thermoacoustic refrigerator device*

II.3.1 Thermoacoustic refrigerator parts

II.3.1.1 Speaker

Speaker is the source of acoustic waves which is also known as Acoustic driver in figure II-5. The acoustic driver produces sound waves. Sound waves are produced due to vibration of flexible cone or diaphragm. The diaphragm is made up of plastic, paper, metal etc. and narrow end of the coil is attached to the coil which produces sound name as voice coil. The voice coil contains two magnets namely permanent magnet and electromagnet. The audio signal transmits or travels in the form of waves it may be transverse and longitudinal. These waves are further travels through stack. A higher performance of speaker leads to higher performance of whole refrigerator system. [60]

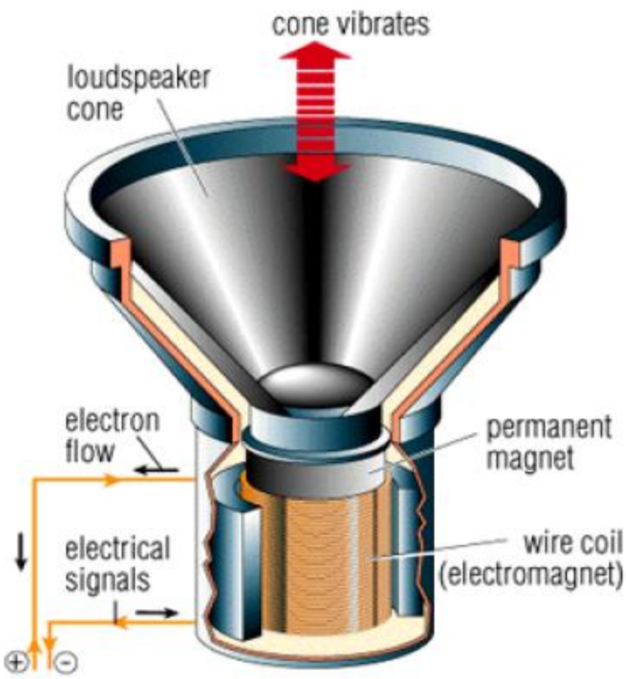


Figure II-5: *thermoacoustic refrigerator speaker*

II.3.1.2 Resonator

The purpose of the resonator is to contain the working fluid in a thermo acoustic refrigerator and create resonance in it at applied frequency. The shape, length and losses are important parameters for designing the resonator. Length of the resonator is determined by resonance frequency and minimum losses at wall of the resonator. Resonators are generally of two types: [66]

II.3.1.2.1 Quarter wavelength Resonator

Quarter wavelength resonators shown in figure II-6 are made with tubes by sealing one end and making the length approximately one quarter of the desired resonant frequency wavelength. The open end of the tube is simulated by attaching a large volume to the end. This large volume creates the boundary condition of zero pressure at the end, causing the end of the tube to be a pressure node and velocity anti-node while the beginning of the resonator is approximately a velocity node and a pressure anti-node. [66]

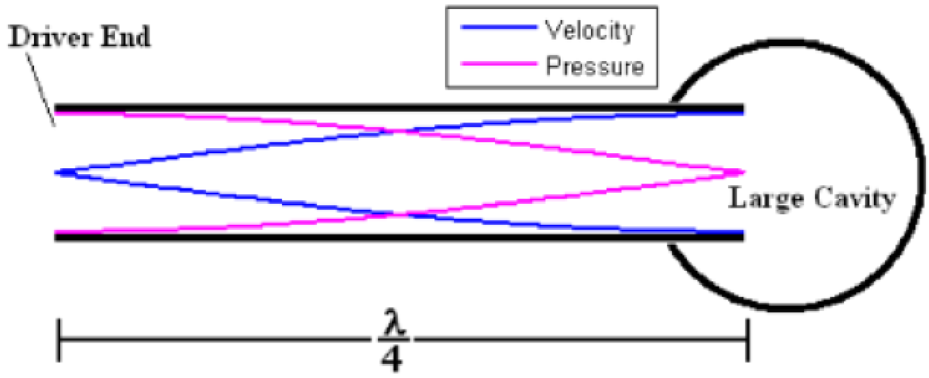


Figure II-6: *Quarter wavelength Resonator*

II.3.1.2.2 Half wavelength resonator

Half wavelength resonators shown in figure II-7 are roughly a long tube that is closed at the end. The closed end means that the gas inside the resonator cannot move, creating a velocity node and pressure anti-node. The driver at the beginning of the tube also creates a velocity node and pressure anti-node, causes the natural frequency of such a cavity to be half the acoustic wavelength. [66]

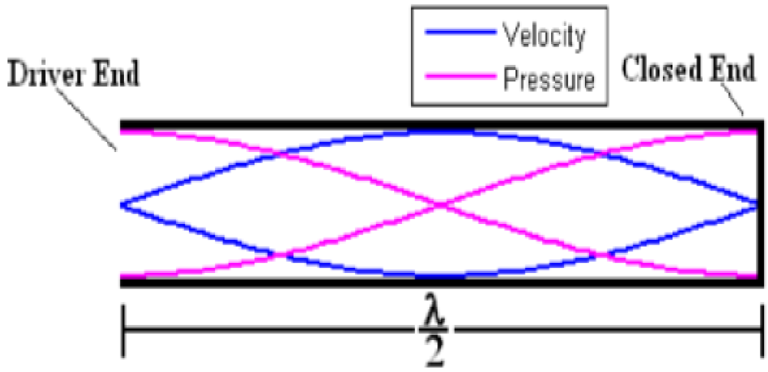


Figure II-7: *Half wavelength Resonator*

II.3.1.3 Stack

Stack is the element where thermo acoustic effect happens. It is the most delicate piece of the plan in thermo acoustic refrigerator since little change in measurements of the stack can lead enormous contrast in its performance. There are two sorts of stack arrangement parallel plate stack and winding stack. The another thought while planning stack will be stack thickness must be diminish so that there must no blockage in the stack. On the off chance that blockage would present, at that point the acoustic wave would not go through the stack and in the event that the thickness is excessively thick, at that point there would be development of vortexes close to the stack. [67]

II.3.1.3.1 Stack geometry

Extensive research has been reported on stack geometry, the most frequent geometries are spiral, parallel, honeycomb and pin array later is most efficient geometry but manufacturing difficulties associated with pin array structure it is abandoned by researchers. However, pin array structure provides effective heat interaction between working gas and solid walls, the final choice base on manufacturing difficulties and conversion efficiency. The next efficient geometry to pin array is parallel plate geometry utilized by researcher's group with favorable results. Figure II-8 shows various stack geometries utilize by researcher's during their studies, spiral stack, parallel, pin array [62], most of the studies carried out with parallel and pin array structure. [12]

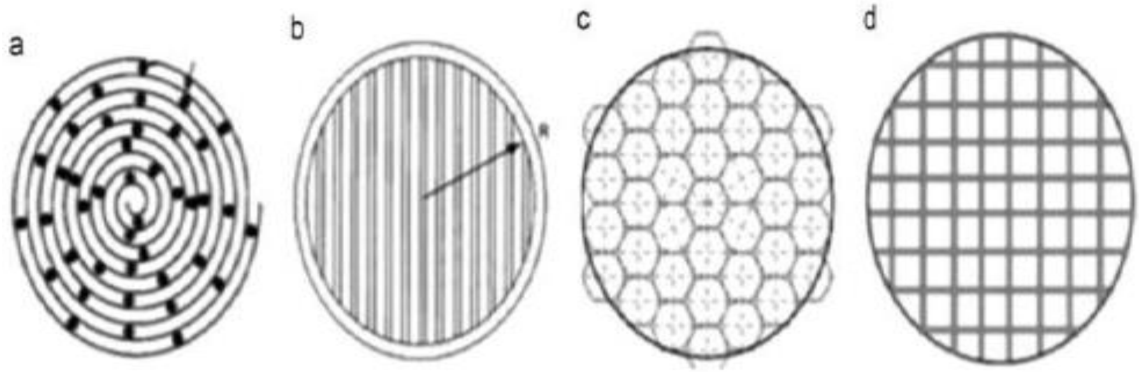


Figure II-8: *(a) spiral, (b) parallel, (c) honeycomb, (d) pin array*

II.3.1.4 Heat Exchangers

Heat exchanger is a component or a device which is use to transfer thermal energy between fluids (figure II-9), it may be two or more than two. There is no external heat or work interaction in heat exchanger. The working of heat exchanger is to remove the heat from high temperature source.

In thermoacoustic refrigerator the heat exchanger is used to remove the heat from the stack so that temperature is maintain as per requirement. In thermo acoustic refrigerator two heat exchangers are required one work as hot heat exchanger and other work as cold heat exchanger. [67]

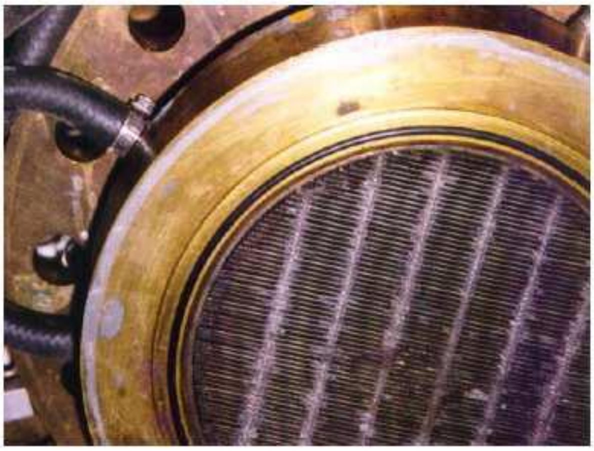


Figure II-9: *Heat Exchangers*

II.3.1.5 Working gas

To dominate the viscous effects in thermoacoustic working gas should have lower Prandtl number, as it is the ratio of viscous effects to thermal effects, lower Prandtl number ensure that more thermal effects within the thermoacoustic device. The working gasses with a higher ratio of specific heat are favorable as the speed of sound reaches its peak. The inert gasses helium, argon, and nitrogen are promising working fluid in thermoacoustic devices as its satisfies the criteria of both higher specific heat ratio and lower Prandtl number, another advantage with inert gasses is that no hazardous effects on the environment. Amongst all inert gasses, the highest sound velocity has achieved in helium. The higher thermal conductivity of working gasses, also beneficial as it enhances heat interaction between oscillating gas within the resonator tube and stack surface results in improvement in cooling effect produced. [63] Other heavy inert gasses i.e. argon, xenon and nitrogen blended to improve the heat interaction but blending should be optimized otherwise these reduce the cooling power Tijani et.al. [64] Used a mixture of Helium-Xenon, Helium-Krypton, and Helium-Argon and Insu et. al [65]. Most of the researchers found air as working fluid for experimentation satisfying criteria of operating condition and design. [12]

II.4 Comparison of thermoacoustic refrigeration system with other refrigeration systems

Apart from vapor compression devices, there are several other ways to provide cooling and refrigeration. Although none of these are currently as versatile as a Vapor Compression Systems but some of these systems hold a high possibility of replacing the pollution causing Vapor Compression Systems. Comparison with various systems is as follows [61]

A. Type of Refrigerant:

The Absorption Refrigeration uses a binary mixture of refrigerant and absorbent like Water/ammonia or LiBr/water. The Adsorption system uses natural refrigerants like water, ammonia or alcohol. Thermo-electric and Thermoacoustic Refrigeration Systems do not use any refrigerant.

B. Working Cycle:

Vapor Absorption Refrigeration is a two-stage process. The vapor refrigerant is absorbed in a binary solution which then regenerates the refrigerant on heating externally. It is cooled in the condenser to the required pressure level and the cycle repeats. Much like the Vapor Compression Refrigeration Systems the Adsorption Systems are also based on withdrawing heat from surroundings during an evaporation process. Thermo-electric System is based on the Peltier Effect wherein an electric current passing through a junction of two materials will cause a change in temperature. The Thermoacoustic Refrigeration System is powered by either a heat engine running on waste heat or an electric source. Due to compression and expansion of air packets heat transfer across two mediums is made possible.

	Thermoacoustic refrigerator	**Thermo-electric**	**Vapor Absorption Refrigeration**
Type of refrigerant	do not use any refrigerant	do not use any refrigerant	• mixture of refrigerant and absorbent like Water/ammonia or LiBr/water • Water, ammonia or alcohol
Working Cycle	• powered by either a heat engine running on waste heat or an electric source Due to compression and expansion of air packets heat transfer across two mediums is made possible	• based on the Peltier Effect wherein an electric current passing through a junction of two materials will cause a change in temperature	two-stage process: • the vapor refrigerant is absorbed in a binary solution which then regenerates the refrigerant on heating externally • It is cooled in the condenser to the required pressure level and the cycle repeats

Table II-1:*Comparison of thermoacoustic refrigeration system with other refrigeration systems*

II.4.1 Advantages, Inconvenient

II.4.1.1 Advantages

- The device can be produced and operated cheaper than the traditional vapor compression cooler due to:
- Mechanical simplicity and very few numbers of the components.
- No need for lubricants since there is only one moving part, which is the loudspeaker.
- No expensive components.
- Use of cheap and readily available gases (air or helium).
- Lower life-cycle cost.
- ❖ **Environmental Friendliness**

  The international restriction on the use CFC (chlorofluorocarbon) and skepticism over the replacements of CFC, gives thermo acoustic devices a considerable advantage over traditional refrigerators. The gases used in these devices are totally harmless to the ozone and have no greenhouse effect. It is expected that in the close future regulations will be tougher on the greenhouse gases. [66]

II.4.1.2 Inconvenient

- Efficiency: Thermo acoustic refrigeration is currently less efficient than the traditional refrigerators.
- Talent bottleneck: There are not enough people who have expertise on the combination of relevant disciplines such as acoustics, heat exchanger design etc.
- Lack of suppliers producing customized components.
- Lack of interest and funding from the industry due to their concentration on developing alternative gases to CFCs.
- Another major problem of TAR is that it is either fully on or off. [66]

Advantages	Inconvenient
• The device can be produced and operated cheaper than the traditional vapor compression cooler • Mechanical simplicity • No need for lubricants • Lower life-cycle cost. • Environmental Friendliness The gases used in these devices are totally harmless to the ozone and have no greenhouse effect	• Less efficiency • Talent bottleneck • Lack of suppliers • Lack of interest and funding from the industry • major problem of TAR is that it is either fully on or off

Table II-2: *Summary of advantages, inconvenient of thermoacoustic refrigerator*

II.5 Conclusion

In this chapter theoretical studies and working principle of TAR is presented, most components such as loudspeaker, resonator, stack, heat exchanger and working fluid are explained as well as the basic laws such as thermodynamics first and second law, ideal gases, COP law. Comparison with other refrigeration systems such as absorption vapor compression and thermo-electric is done. At the end of the chapter a summary of the advantages and inconvenients of the TAR is presented.

Chapter III: The studied case

III.1 Introduction

Thermodynamics is a science which is based upon the general laws of nature which govern the conversion of heat into mechanical work and vice-versa. In terms of classification, thermodynamics is classified as part of classical mechanics, which is a branch of physics where we study the behavior of a system by considering only the large-scale response (macroscopic properties). This is unlike modern thermodynamics which attempts to explain the behavior of matter from a microscopic or atomic viewpoint. Thermodynamics this science is concerned with a lot of domains such as thermal energy storage, electricity production (steam and gas turbine) …etc. and one of these domains is one of the most important in the modern time which is the refrigeration system. [73]

III.2 Refrigeration systems

III.2.1 Vapor compression refrigeration cycle

A basic circuit of vapor compression refrigeration is shown in Figure III-1. In the evaporator some of the refrigerant changes phase from liquid to vapor as a result of heat transfer from the cold region to the refrigerant, the refrigerant is then compressed After the compression of vapor, it is condensed and then expanded to a low temperature through a valve in which the process is essentially at constant enthalpy. [74]

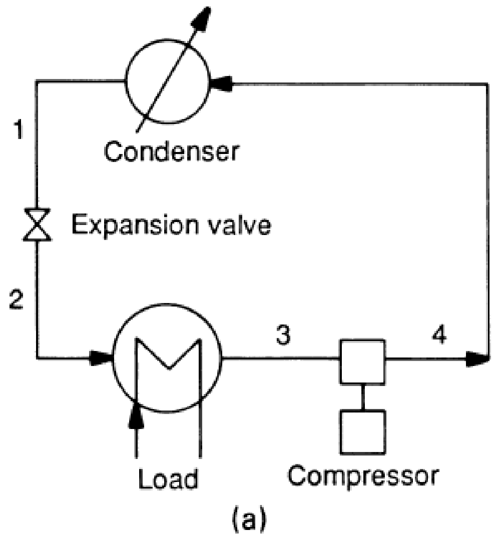


Figure III-1: *vapor compression refrigeration*

III.2.2 Absorption refrigeration cycle

The absorption refrigeration system consists of two cycles, the regeneration cycle and the absorption cycle. During the regeneration cycle, heat is supplied to the system to generate ammonia vapor, which then passes to the condenser to be condensed and stored as ammonia liquid, in an insulated receiver. During the absorption cycle, ammonia liquid vaporizes in the evaporator, and hence, produces cooling effect, and flows back to be absorbed by the weak solution, in the absorber.[76]

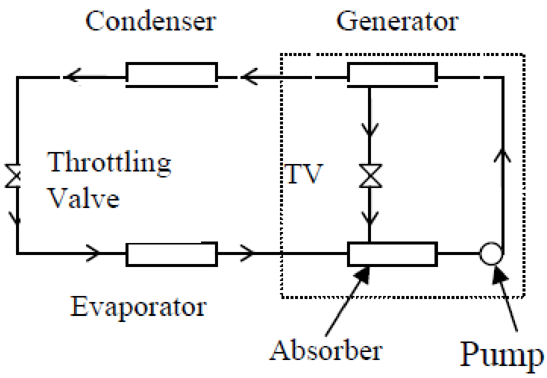


Figure III-2: *Absorption cycle*

III.2.3 Brayton refrigeration cycle

The Brayton refrigeration cycle is the reverse of the closed Brayton power cycle. A schematic of the reversed Brayton cycle is presented in Figure III-3. The refrigerant gas, which may be air, enters the compressor, and is compressed. The gas is then cooled. and expanded. [75]

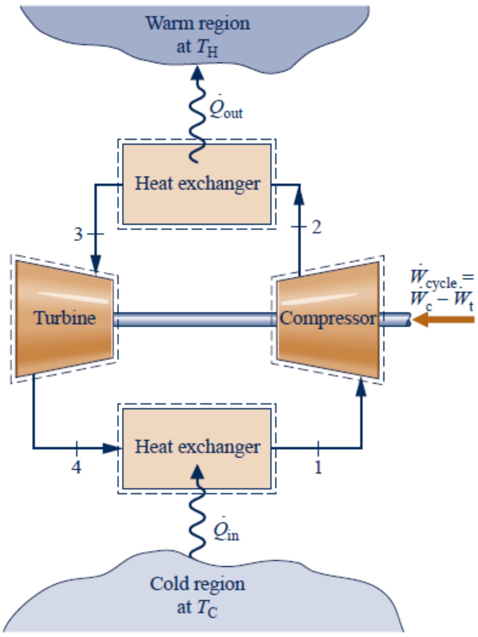


Figure III-3: *Brayton refrigeration cycle.*

III.2.4 Thermoacoustic refrigeration cycle

The cycle by which heat transfer occurs is similar to the Stirling cycle. Figure III.4 traces the basic thermoacoustic cycle for a packet of gas. The packet of gas is compressed and moves to the left. When the gas packet is at maximum compression, the gas ejects the heat back into the stack since the temperature of the gas is now higher than the temperature of the stack. As the gas packet moves back towards the right, the sound wave expands the gas temperature decrease. Finally, the packets of gas reabsorb heat from the cold reservoir to repeat the heat transfer process. [66]

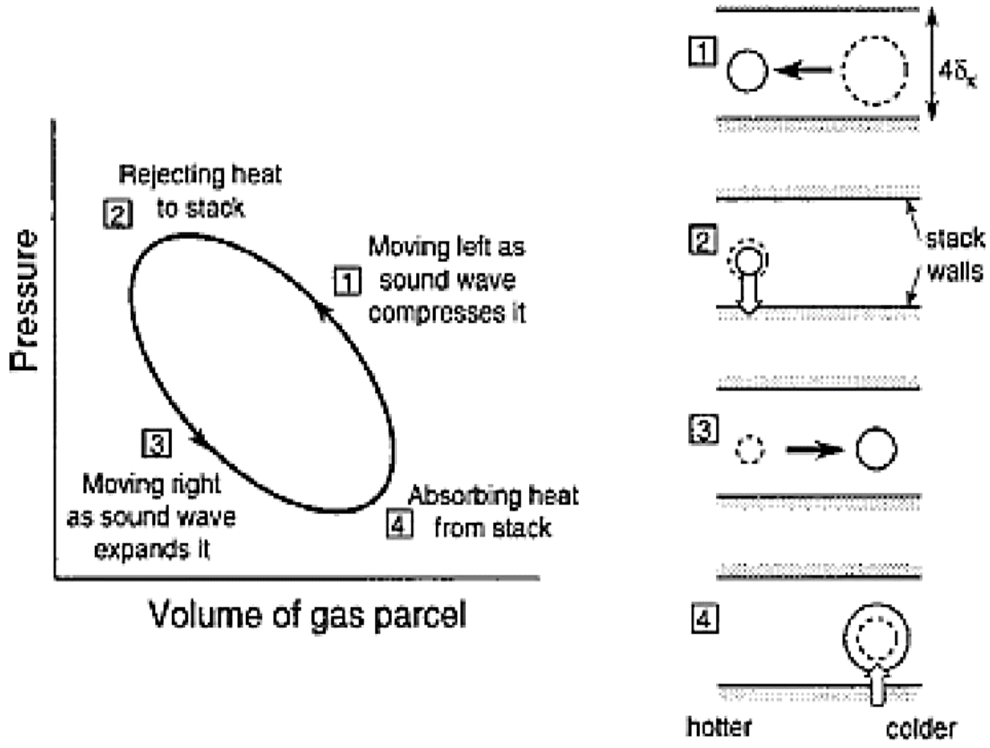


Figure III-4: *Thermo-acoustic refrigeration cycle*

III.3 Deltaec description

DeltaEC, the abbreviation of Design Environment for Low-Amplitude Thermo-Acoustic Energy Conversion, is a computer program made by Los Alamos National Laboratory that can calculate details of how thermoacoustic equipment performs. Input data can be modified or entered via DeltaEC's user interface. Results can be examined via the user interface... [72]

DeltaEC contains a user interface which is shown in Figure III-5, this user interface has all of the segments of a model strung together. Each segment has a number of variables within it which describe the physical characteristics of the segment. These variables are input by the user.

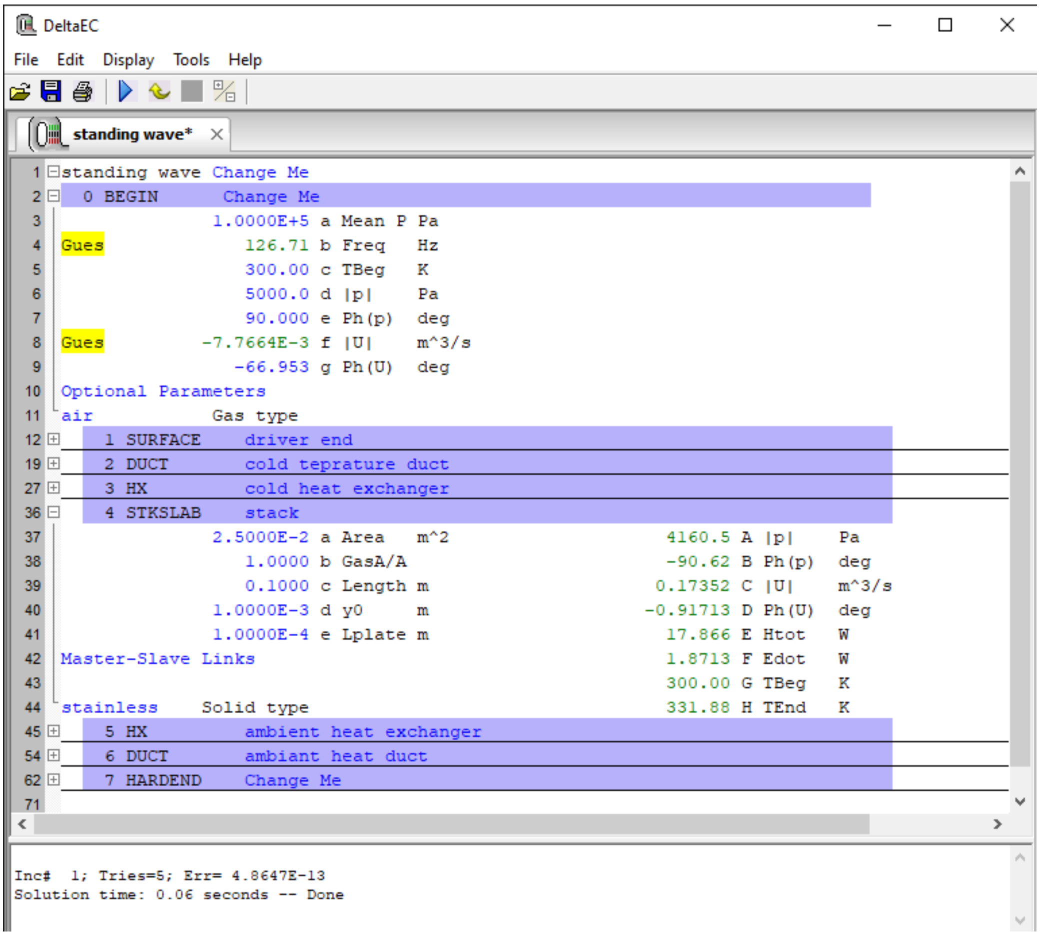


Figure III-5: DELTAEC User Interface

III.4 Modeling in DELTAEC

DeltaEC numerically integrates the momentum, continuity, and energy equations and solves these equations. Models are made by the numerical integration carried out across the created network of segments. [70]

III.5 DELTAEC model

Models are built from a combination of segments strung together. The studied thermoacoustic refrigerator is modeled using 14 segments in DELTAEC. The length of the device is 5m.

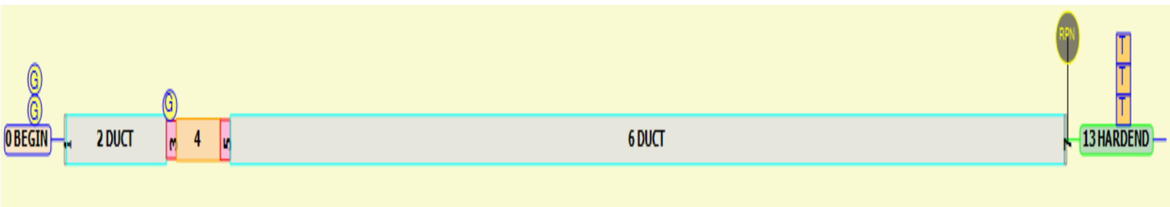


Figure III-6: *schematic view of studied TAR using DeltaEC*

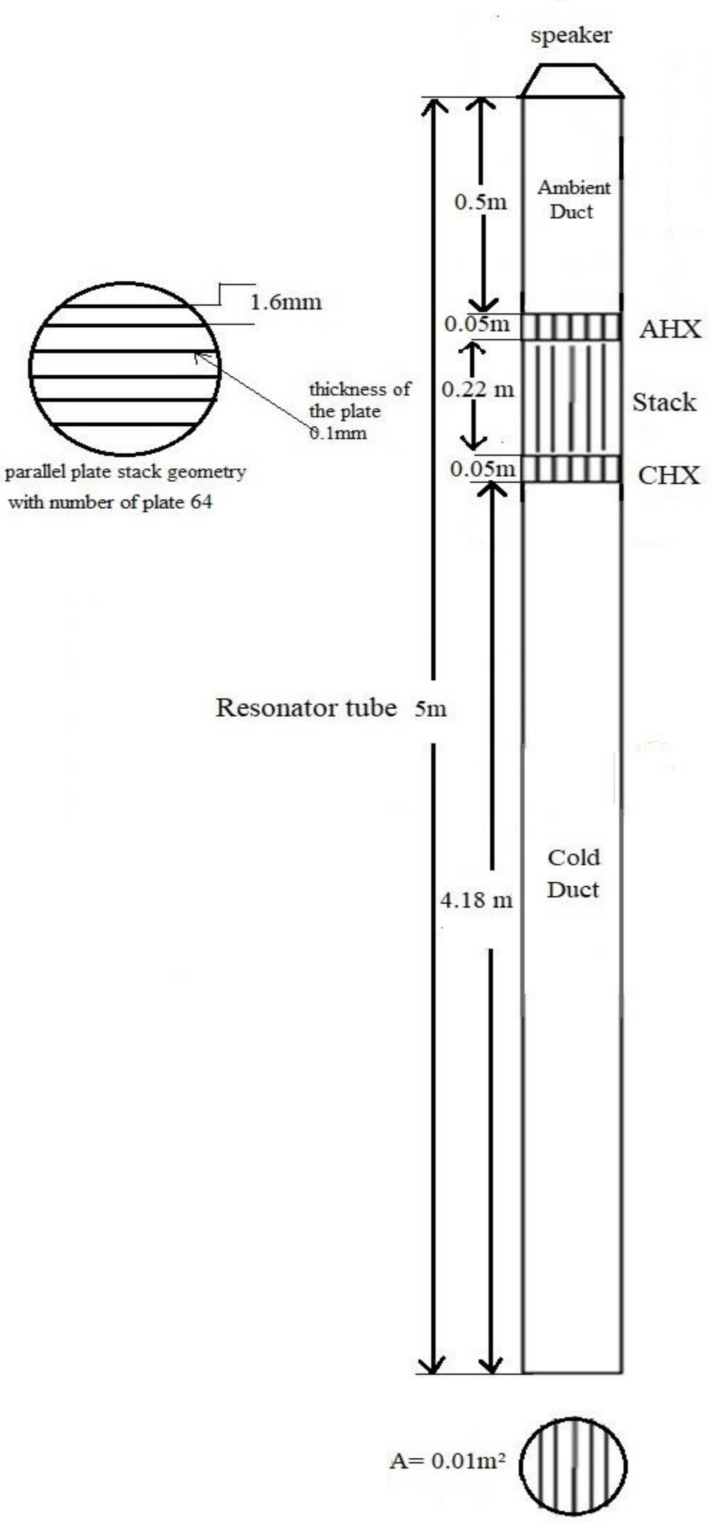


Figure III-7: *Studied TAR geometry*

the geometry of the TAR as presented on figure III-7 has a total length of 5m and a cross sectional area of 0.01m², 0.5m of length for the ambient duct then 0.05m for the AHX, then a stack of parallel plate geometry with a length of 0.22m and space between plates 1.6mm and 0.1mm of thickness with number of plate 64 , the next part is the CHX with the same length of AHX, at the end the cold duct with a length of 4.18m

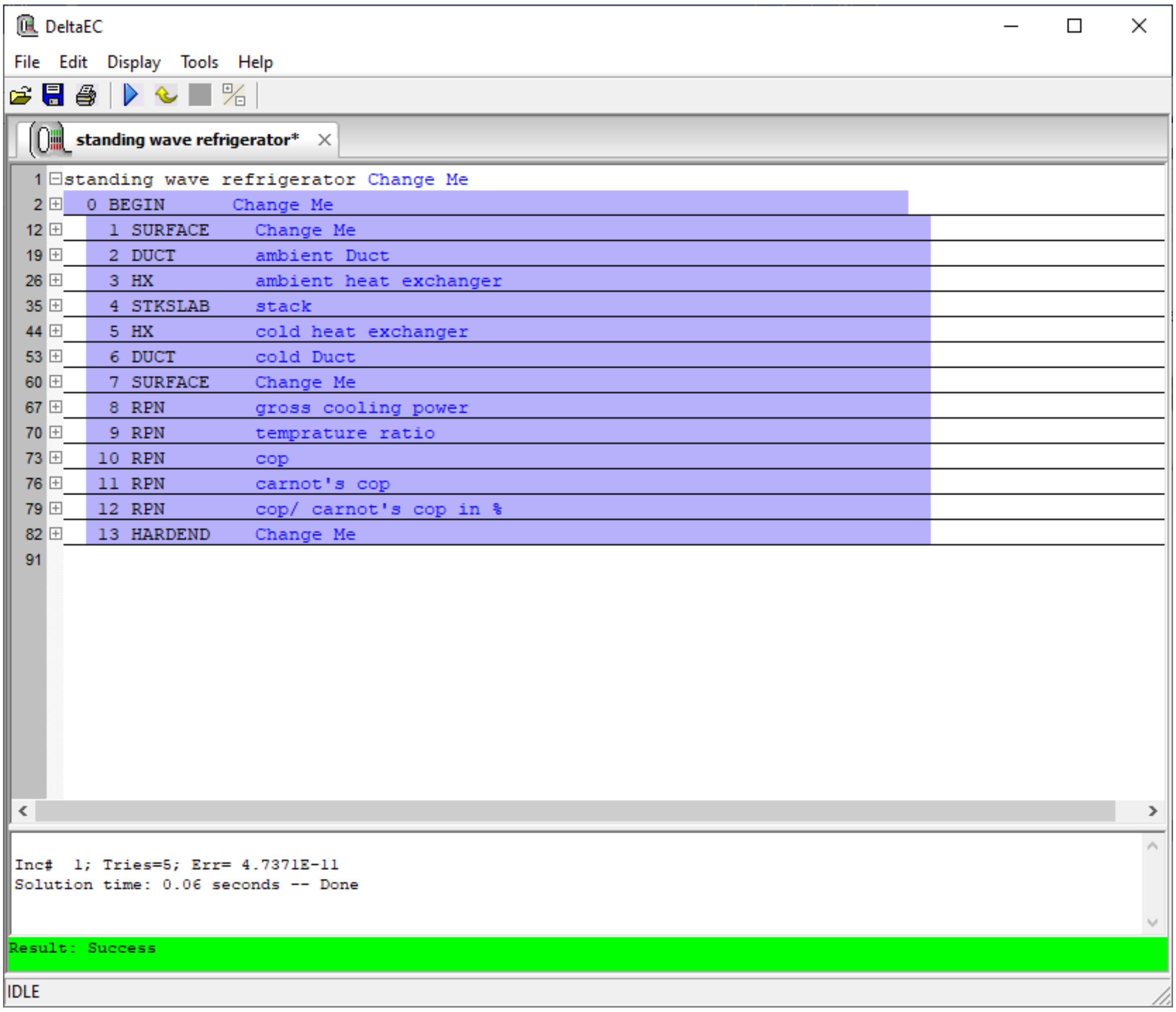


Figure III-8:*DeltaEc model*

❖ The first segment is the **BEGIN** segment which contains the parameters of the thermoacoustic refrigerator. This segment is shown in Figure III-9.

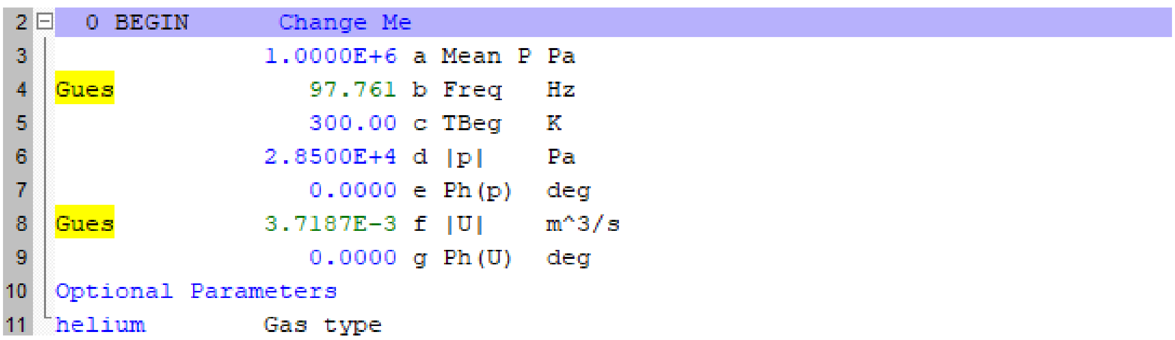


Figure III-9: *Segment 0 modeling parameters for the thermoacoustic refrigerator*

This segment contains the mean pressure for the thermoacoustic refrigerator, which is equal to the atmospheric pressure in the line labeled by 3. It indicates also the driving frequency which is 97.761 Hz and the starting temperature throughout the refrigerator of 300K in lines 4 and 5. Lines 6 and 7 represent the magnitude and phase of the pressure input to the back of the speaker diaphragm. Since the speaker is electrically driven only, these parameters are left zero. Lines 8 and 9 contain the magnitude and phase of the volumetric flow rate at the back of the speaker. Because the volumetric flow rate is dependent on the effort variables, this parameter is labeled as guess and is part of the solution in DeltaEC. Also, it is shown in Figure III-9 that the gas type is selected to be helium. [70]

❖ The second segment is the **SURFACE**

```
12    1 SURFACE     Change Me
13                 1.0000E-2 a Area    m^2                  2.8500E+4 A |p|     Pa
14                                                             0.0000 B Ph(p)   deg
15                                                          3.7101E-3 C |U|     m^3/s
16                                                             0.0000 D Ph(U)   deg
17                                                             52.991 E Htot    W
18  ideal          Solid type                                  52.869 F Edot    W
```

Figure III-10: *segment 1" SURFACE"*

The solid material surface area exposed to oscillating pressure is equal to $0.01m^2$

❖ Third segment **DUCT**

```
19    2 DUCT          ambient Duct
20                 1.0000E-2 a Area    m^2     Mstr         2.7207E+4 A |p|     Pa
21                   0.35449 b Perim   m       2a            -0.36312 B Ph(p)   deg
22                    0.5000 c Length  m                    5.1981E-2 C |U|     m^3/s
23  Master-Slave Links                                        -86.252 D Ph(U)   deg
24  Optional Parameters                                        52.991 E Htot    W
25  ideal          Solid type                                  50.694 F Edot    W
```

Figure III-11: *segment 2 DUCT*

Figure III-11 above shows the first duct segment which is next in the series. This segment models the part of the resonator between the speaker and the stack. Line 20 is the cross-sectional area of this tube while lines 21 and 22 are the perimeter of the cross section and length of the tube respectively.

❖ Fourth segment **Heat Exchanger (HX)**

```
26  3 HX        ambient heat exchanger
27              1.0000E-2 a Area    m^2        2.6710E+4 A |p|    Pa
28                 0.6000 b GasA/A            -0.28192 B Ph(p)   deg
29              5.0000E-2 c Length m           5.5238E-2 C |U|    m^3/s
30              8.0000E-4 d y0     m            -86.806 D Ph(U)   deg
31 Gues           -82.991 e HeatIn W             -30.00 E Htot    W
32 Master-Slave Links                             44.720 F Edot    W
33 Possible targets                               300.00 G GasT    K
34 ideal          Solid type                      299.39 H SolidT  K
```

Figure III-12: *segment 3 heat exchanger HX*

The heat exchanger was modeled as shown in Figure III-12. In Line 27 is the cross-sectional area of this heat exchanger and in line 28, the ratio of gas to solid in the cross section. Line 29 gives the length of the heat exchanger, line 30 the spacing between layers of the heat exchanger, and line 31 the heat Q added to the thermoacoustic gas, positive Q indicates heat flow from solid to gas.

❖ Fifth segment **stack**

```
35  4 STKSLAB     stack
36  Same  3a 1.0000E-2 a Area    m^2        2.4474E+4 A |p|    Pa
37                 0.7240 b GasA/A             0.19451 B Ph(p)   deg
38                 0.2200 c Length m           7.1226E-2 C |U|    m^3/s
39              8.0000E-4 d y0     m            -88.248 D Ph(U)   deg
40              1.0000E-4 e Lplate m             -30.00 E Htot    W
41 Master-Slave Links                             23.684 F Edot    W
42                                                300.00 G TBeg    K
43 stainless      Solid type                      272.88 H TEnd    K
```

Figure III-13: *segment 4 stack (STKLAB)*

The stack was modeled as shown in Figure III-13. In Line 36 is the cross-sectional area of this stack and in line 37, the ratio of gas to solid in the cross section. Line 38 gives the length of the stack, line 39 the spacing between layers of the stack, and line 40 the thickness of the film which the stack is constructed out of.

The next segment in the model is the cold heat exchanger, the cold heat exchanger was modeled in the same way as the ambient heat exchanger.

```
44  5 HX        cold heat exchanger
45              1.0000E-2 a Area    m^2        2.3817E+4 A |p|    Pa
46                 0.6700 b GasA/A             0.36263 B Ph(p)   deg
47              5.0000E-2 c Length m           7.4479E-2 C |U|    m^3/s
48              8.0000E-4 d y0     m            -88.527 D Ph(U)   deg
49                 30.000 e HeatIn W         -1.4211E-14 E Htot    W
50 Master-Slave Links                             17.181 F Edot    W
51 Possible targets                               272.88 G GasT    K
52 ideal          Solid type                      273.04 H SolidT  K
```

Figure III-14:*Segment modeling the heat input to the cold side of the refrigerator from the outside*

The cold duct was modeled with same way as the hot duct just the difference in length

```
 6 DUCT          cold Duct
            1.0000E-2 a Area    m^2      Mstr        2.7024E+4 A |p|    Pa
              0.35449 b Perim   m        6a            -179.39 B Ph(p)  deg
               4.1800 c Length  m                    7.4694E-6 C |U|    m^3/s
Master-Slave Links                                     -179.39 D Ph(U)  deg
Optional Parameters                                 -1.4211E-14 E Htot   W
ideal           Solid type                             0.10093 F Edot   W
```

Figure III-15: *the segment modeling cold duct*

The Next segment is the surface was modeled in the same way as the first surface

```
 7 SURFACE       Change Me
            1.0000E-2 a Area    m^2                  2.7024E+4 A |p|    Pa
                                                       -179.39 B Ph(p)  deg
                                                    1.6208E-17 C |U|    m^3/s
                                                       -96.867 D Ph(U)  deg
                                                   -1.4211E-14 E Htot   W
ideal           Solid type                          2.8486E-14 F Edot   W
```

Figure III-16:*second surface segment*

- Sixth segment RPN

```
 8 RPN          gross cooling power
                0.0000 a G or T                    47.181    A watt
5e 5F +
 9 RPN          temprature ratio
                0.0000 a G or T                    0.91197 A ratio
5H 3H /
 10 RPN         cop
                0.0000 a G or T                    0.89241 A cop
8A 1F /
 11 RPN         carnot's cop
                0.0000 a G or T                    10.360    A carnot'
5H 3H 5H - /
 12 RPN         cop/ carnot's cop in %
                0.0000 a G or T                    8.6143    A in %
10A 11A / 100 *
```

Figure III-17: *RPN segments*

RPN is an abbreviation of Reverse Polish Notation (also called Postfix Notation), which is a way of writing algebraic expressions without the use of parentheses or rules of operator precedence (Brown, 2005).

So basically, DELTAEC uses encoding technique which is algebraic parenthesis-free expressions and the user can define RPN segments to determine the required output parameters like COP of the refrigeration and other desired parameters. [76]

❖ Seventh segment **Hardend**

```
   13 HARDEND         Change Me
Targ                  0.0000 a R(1/z)                   2.7024E+4 A |p|     Pa
Targ                  0.0000 b I(1/z)                     -179.39 B Ph(p)   deg
Targ                  0.0000 c Htot    W                1.6093E-17 C |U|     m^3/s
                                                          -89.373 D Ph(U)   deg
                                                        1.4211E-14 E Htot    W
                                                       -7.9031E-17 F Edot    W
                                                       -3.7112E-20 G R(1/z)
                                                        1.0211E-16 H I(1/z)
```

Figure III-18: *segment 13 HARDEND*

The final segment is titled the HARDEND. This segment is a wall at the end of the cold duct segment which terminates the refrigerator. It is shown in Figure III-18 above. This segment contains three variables which are the reciprocals of the real and imaginary parts of the acoustic impedance and the total energy flow. The first two variables are set to zero indicating that there is infinite acoustic impedance at this point. [70]

III.6 Numerical simulation

Numerical simulation is used to study the following elements and results are showed in chapter 4.

First, the position of the stack that been changed in order to know their effect on the temperature difference and the coefficient of performance. The mean pressure and the acoustic pressure in the resonator are fixed at 10 bar and 0.285 bar, respectivelly and the position of stack change from the distance of 0.36m to 1.96m. This study is performed to find the optimal position of the stack in the standing wave thermoacoustic refrigerator with a half-wave length.

Second, after choosing the optimal position of the stack, the next step is to study the effect of mean pressure on the temperature difference and the coefficient of performance and the increase of mean pressure is from 6 bar to 14 bar.

Third, after choosing the optimal position of the stack and fixing mean pressure the acoustic pressure is studied in the interval of pressure between 0.2 bar to 0.8 bar.

Finally, the heat power input and the temperature difference are fixed at 82.991W and at 27K, respectively in order to select the optimum gas that maybe used in this model.

III.7 Conclusion

This chapter presented a general description about thermodynamics, common types of refrigeration vapor-compression, absorption, reversed Brayton cycles and thermoacoustic refrigeration. A detailed modeling of the sanding wave thermoacoustic refrigerator using DELTAEC software package is performed. A description about the numerical simulation steps applied on the modeled thermoacoustic refrigerator is done.

Chapter IV: Results and discussion

IV.1 Introduction

In Chapter 3, the studied standing wave thermoacoustic refrigerator is presented and modeled using the thermoacoustic modeling software DELTAEC. In this chapter the results are introduced as followed, first the influence of stack position on temperature difference along the resonator is presented. Then the impact of mean pressure and acoustic pressure on the temperature difference and coefficient of performance (COP) is shown. Finally, the influence of different working fluids on the acoustic pressure and frequency are presented, where temperature difference, mean pressure and heat add to the refrigerator is fixed for all used gases.

IV.2 Results and Discussion

Figure IV-1 shows the acoustic pressure and the acoustic velocity of a standing wave thermoacoustic refrigerator. The speaker at the beginning of the resonator tube creates a velocity node and pressure anti-node at steady state operation. This causes the natural frequency of oscillation with half wavelength of the acoustic pressure and velocity since the both ends are closed. In the standing wave resonator, acoustic pressure and velocity are in phase of 90°, this mean that the maximum pressure occurs at the point of zero velocity and minimum pressure at maximum velocity.

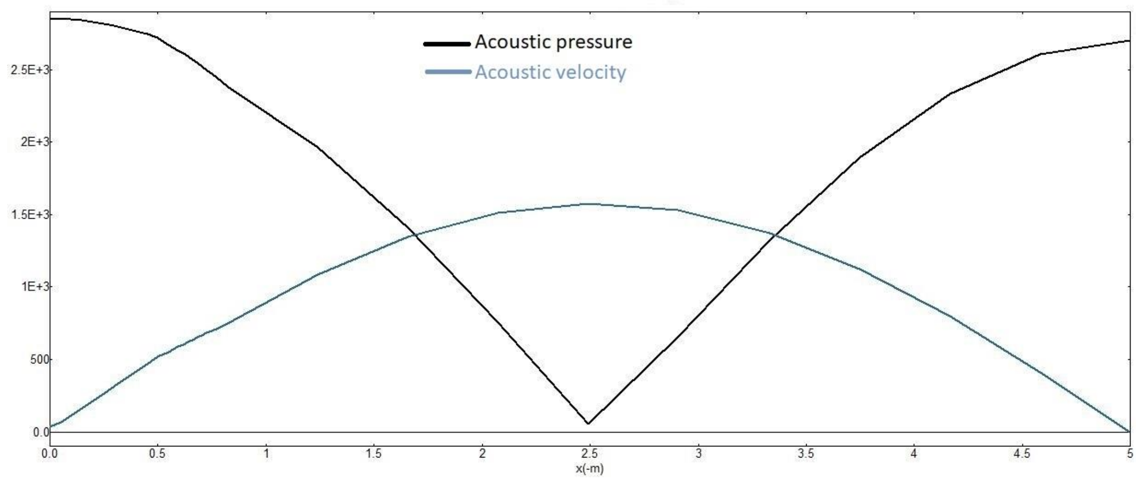


Figure IV-1: *acoustic pressure and velocity distribution in a standing wave thermoacoustic refrigerator*

Figure IV-2 shows the mean temperature plotted along the standing wave thermoacoustic refrigerator. As seen in this figure the temperature difference between the ambient and the cold part of the stack at steady state operation is approximately 27K due to the thermoacoustic effect.

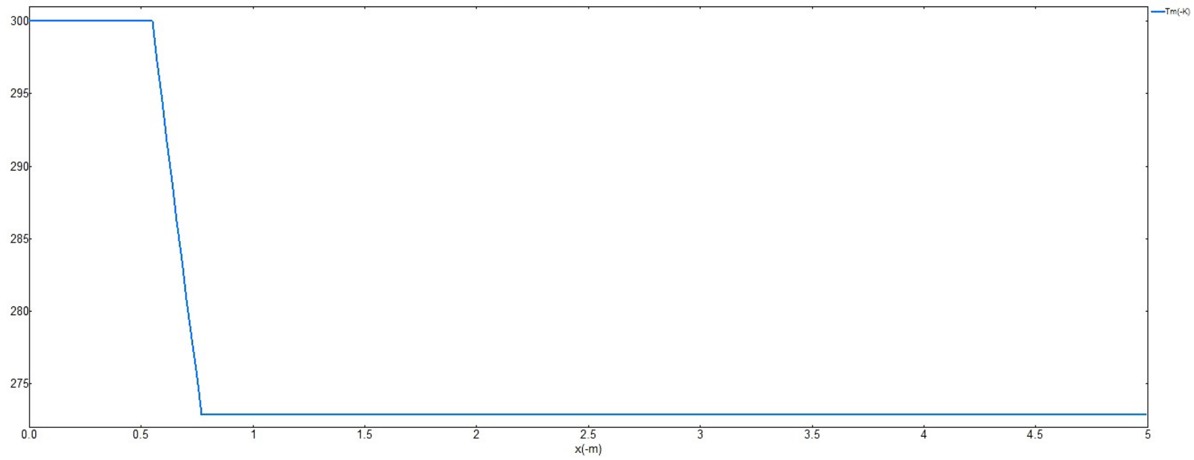


Figure IV-2: *temperature distribution in standing wave thermoacoustic refrigerator*

IV.3 Study of the optimal Stack position

Figure IV-3 shows the variation of temperature difference with respect to the stack position along the resonator tube. An increase in temperature difference between the ambient and the cold part of the stack is observed where it rises from 15 to 26.35K when the stack moves from the position of 0.36m to 0.66m. Then, a decrease is noted until the stack position reach 1.96m. Above this position, the DELTAEC software is not able to predict the thermoacoustic effect of refrigerator.

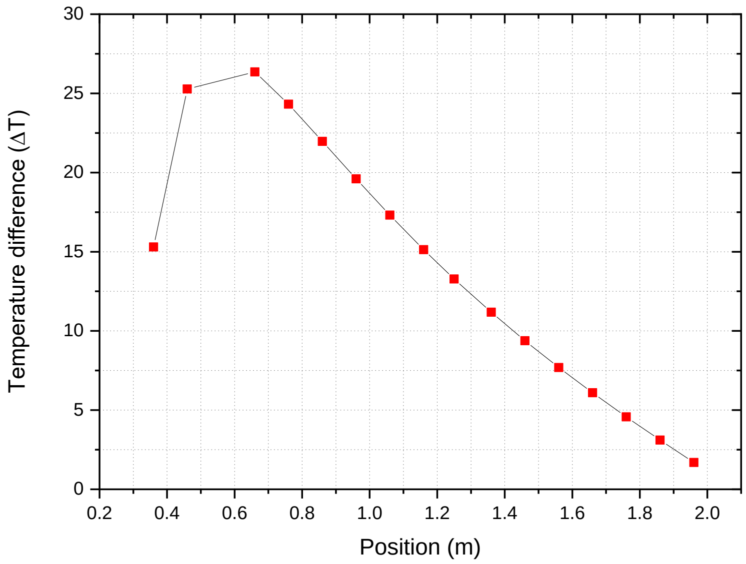


Figure IV-3*: variation of temperature difference with respect of the stack position*

Figure IV-4 shows the variation of the coefficient of performance **COP** with respect to the stack position. A decrease in coefficient of performance from 19.87 to 10.36 is observed, due to the increase of the temperature difference, when the stack position shifts from 0.36m to 0.66m. Then, an increase is noted until the stack position reach 1.96m. Above this position, the DELTAEC software is not able to predict the thermoacoustic effect of refrigerator.

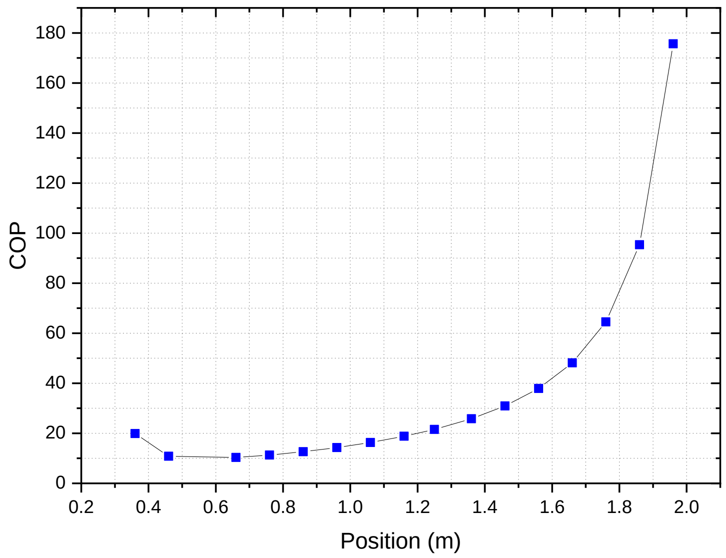


Figure IV-4: *variation of COP with respect of the stack position*

Based on these results the optimal position of the stack is 0.66m because at this position thermoacoustic refrigerator produce the maximum value of the temperature difference.

IV.4 Study of the mean pressure inside the resonator tube

Figure IV-5 shows the variation of temperature difference with respect to the mean pressure. It is observed that the increases in the mean pressure inside the resonator tube from 6 to 14bar leads to a decrease in the temperature difference from the value of 31.6 to 18.62K.

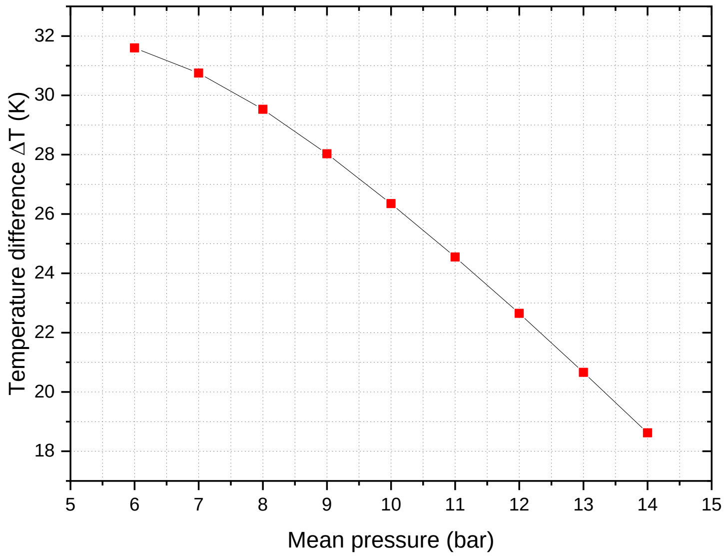


Figure IV-5: *Variation of temperature difference with respect to variation of the mean-pressure*

Figure IV-6 shows the variation of coefficient of performance COP with respect of the mean-pressure variation. It is observed that when the mean pressure increases it leads to an increase the coefficient of performance COP due to the decrease of the temperature difference.

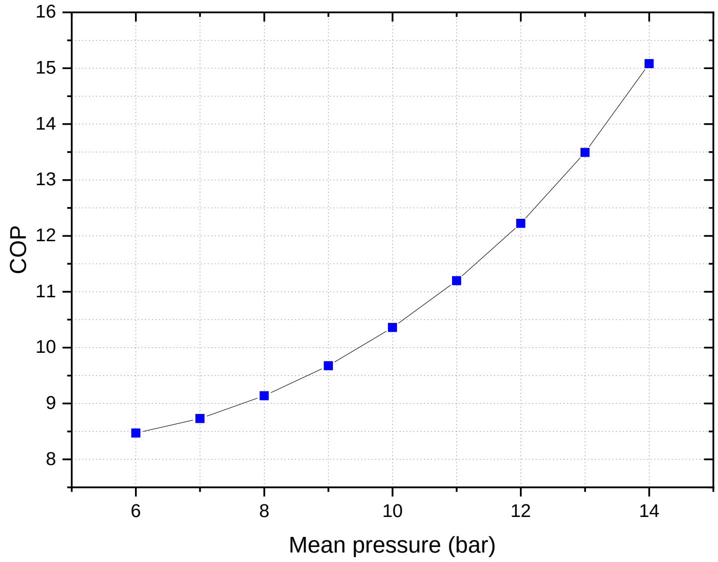


Figure IV-6: *variation of COP with respect of the mean-pressure variation*

IV.5 Study of the acoustic pressure inside the resonator tube

Figure IV-7 shows the variation of temperature difference with respect to the acoustic pressure inside the resonator tube. It is observed that the rise of acoustic pressure from 0.2 to 0.8bar leads to an increase the temperature difference from 14.51 to 36.73K.

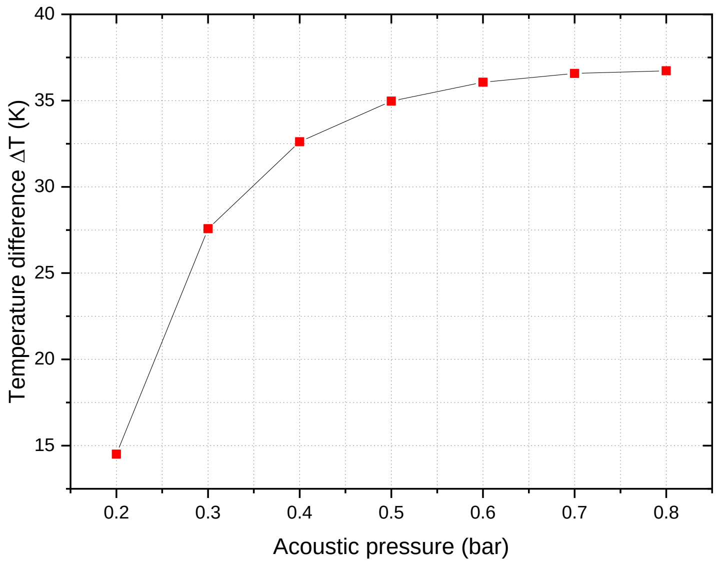


Figure IV-7:*Variation of temperature difference with respect of variation of the acoustic pressure*

Figure IV-8 shows the variation of coefficient of performance COP with respect of the acoustic pressure variation. It is observed that when the acoustic pressure increases the coefficient of performance COP decrease.

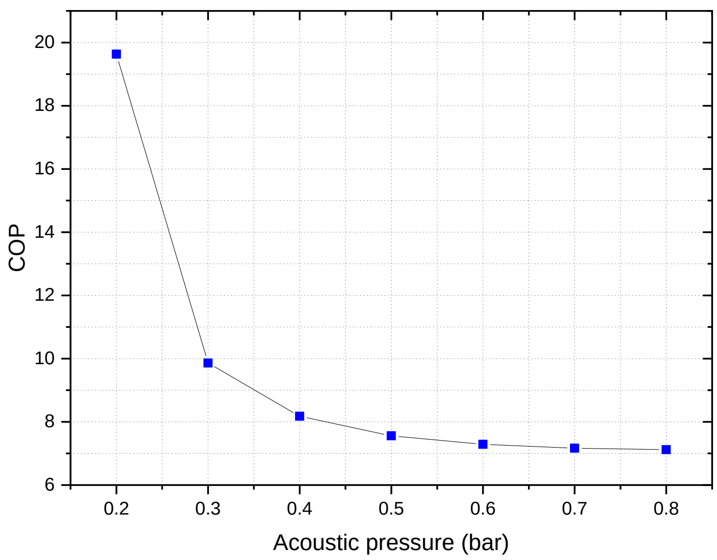


Figure IV-8: *Variation of coefficient of performance COP with respect of the acoustic pressure*

IV.6 Effect of different gases on refrigeration performances

Figure IV-9 shows the variation of acoustic pressure for different gas. In order to produce the same temperature difference in the thermoacoustic refrigerator. It is observed that helium gas needs the minimum value of acoustic pressure of 0.285bar which is the optimum gas compared to NeXe and HeXe that consumes the max acoustic pressure of 0.986bar. Other gases like Air, HiumidAir and nitrogen have the same acoustic pressure. For neon the value of acoustic pressure is 0.46bar and for the HeAr gas the acoustic pressure is 0.64bar.

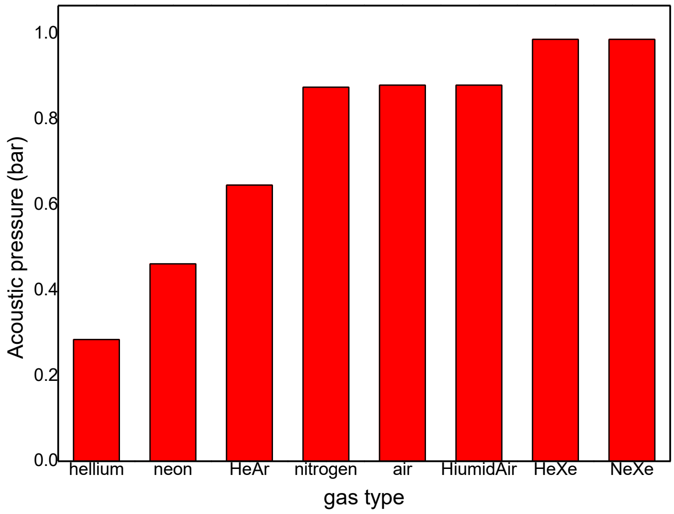


Figure IV-9: *Variation of acoustic pressure for different gases*

Figure IV-10 shows the variation of frequency for difference gases, In order to produce the same temperature difference in the thermoacoustic refrigerator, it is observed that HeXe gas needs the minimum value of frequency of 17.1Hz compared to helium that needs the max frequency of 97.76Hz, Some gases have the same frequency such as Air and HiumdAir because they have the same value of speed of sound

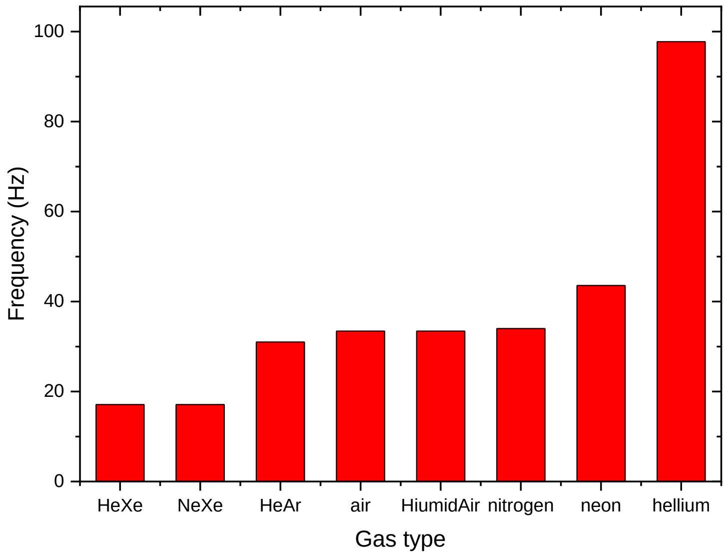


Figure IV-10: *Variation of frequency with gas type*

IV.7 Conclusion

We discussed in this chapter the influence of the stack position, mean pressure and the acoustic pressure on temperature difference and the coefficient of performance with helium as a working fluid, as well as the influence of changing working fluid on the acoustic pressure. The results obtained shows that the stack position and the acoustic and mean pressure have a big influence on the performance of the standing wave thermoacoustic refrigerator.

Conclusion

In this master memoir we present the various important studies that have been made on the thermoacoustic refrigeration system. The theoretical studies and working principle of thermoacoustic refrigerator is presented, as well as most components such as loudspeaker, resonator, stack, heat exchanger and working fluid. A general description about thermodynamics, the basic laws such as first and second law of thermodynamics and common types of refrigeration systemes as vapor-compression, absorption, reversed Brayton cycles and thermoacoustic refrigeration, are explained. A comparison between TAR and other refrigeration systems is done. A detailed modeling of the sanding wave thermoacoustic refrigerator using DELTAEC software is performed.

The results obtained shows that the stack position and the acoustic and mean pressure have a big influence on the performance of the standing wave thermoacoustic refrigerator, and the Helium is the optimum gas for a thermoacoustic refrigerator, the optimum position of stack is at 0.66m along the resonator tube and the best working fluid for the TAR is the Helium, because it required the minimum value of acoustic pressure of 0.285bar and a mean pressure of 10bar.

Bibliography

[1] M. E. Poese, R. W. Smith, S. L. Garrett, R. van Gerwen, and P. Gosselin, “Thermoacoustic refrigeration for ice cream sales,” 6th Gustav Lorentzen Nat. Work. Fluids Conf., pp. 1–8, 2004.

[2] I. Paek, J. E. Braun, and L. Mongeau, “Evaluation of standing-wave thermoacoustic cycles for cooling applications,” Int. J. Refrig., vol. 30, no. 6, pp. 1059–1071, 2007.

[3] R. C. Dhuley and M. D. Atrey, “Investigations on a Standing Wave Thermoacoustic Refrigerator,” pp. 335–342.

[4] S. Humaira, S. Mahmud, and R. Andrew, “Second law analysis of porous thermoacoustic stack systems,” Appl. Therm. Eng., vol. 31, no. 14–15, pp. 2301–2311, 2011.

[5] M. Perier-Muzet, J.-P. Bedecarrats, P. Stouffs, and J. Castaing-Lasvignottes, “Design and dynamic behaviour of a cold storage system combined with a solar powered thermoacoustic refrigerator,” Appl. Therm. Eng., vol. 68, no. 1–2, pp. 115–124, 2014.

[6] S. Sastry, “A Thermoacoustic Engine Refrigerator System for Space Exploration Mission,” no. May, 2011.

[7] P. Saechan, H. Kang, X. Mao, and A. J. Jaworski, “Thermoacoustic Refrigerator Driven by a Combustion-Powered Thermoacoustic Engine – Demonstrator of Device for Rural Areas of Developing Countries,” vol. III, no. 0, pp. 1–6, 2013.

[8] ZOLPAKAR, Nor Atiqah, MOHD-GHAZALI, Normah, AHMAD, Robiah, et al. Performance of a 3D-printed stack in a standing wave thermoacoustic refrigerator. Energy Procedia, 2017, vol. 105, p. 1382-1387.

[9] Jonathan Newman , Bob Cariste , Alejandro Queiruga , Isaac Davis, Ben Plotnick , Michael Gordon , and Sidney San Martín , ”Thermo-acoustic Refrigeration GSET Research Journal” 2006.

[10] KAROKAR, Saurabh, SATPUTE, Rajjat, PARATE, Rohan, *et al.* Design, Fabrication and Analysis of Thermo-Acoustic Refrigeration System–A Review. 2017.

[11] L. K. Tartibu, “Maximum cooling and maximum efficiency of thermoacoustic refrigerators,” Heat Mass Transf., 2015

[12] BANSOD, Prajwal C. et RAUT, Ashish S. Review on Thermoacoustic Refrigeration. International Journal of Innovations in Engineering and Science, 2017, vol. 2, no 3, p. 18-24.

[13] M. E. H. Tijani, J. C. H. Zeegers, and A. T. A. M. De Waele, “Construction and performance of a thermoacoustic refrigerator,” Cryogenics (Guildf), vol. 42, no. 1, pp. 59–66, 2002.

[14] G. Allesina, “An experimental analysis of a stand-alone standing-wave thermoacoustic refrigerator,” pp. 1–9, 2014.

[15] E. C. Nsofor and A. Ali, "Experimental study on the performance of the thermoacoustic refrigerating system," Appl. Therm. Eng., vol. 29, no. 13, pp. 2672–2679, 2009.

[16] N. Yassen, "Impact of Temperature Gradient on Thermoacoustics Refrigerator," Energy Procedia, vol. 74, pp. 1182–1191, 2015.

[17] BABU, K. Augustine et SHERJIN, P. A Critical Review on Thermoacoustic Refrigeration and its Significance. International Journal of ChemTech Research, 2017, vol. 10, no 7, p. 540-552.

[18] PranavMahamuni, Pratik Bhansali, Nishak Shah, Yash Parikh, A Study of Thermoacoustic

Refrigeration System, International Journal of Innovative Research in Advanced Engineering (IJIRAE), 2015, 160-164.

[19] ChannarongWantha, KriengkraiAssawamartbunlue, The Impact of the Resonance Tube on Performance of a Thermoacoustic Stack, Frontiers in Heat and Mass Transfer (FHMT), 2011, 01-08.

[20] A. Namdar, A. Kianifar, and E. Roohi, "Numerical investigation of thermoacoustic refrigerator at weak and large amplitudes considering cooling effect," Cryogenics (Guildf)., vol. 67, pp. 36–44, 2015.

[21] Davis Marx, Xiaoan Mao, Artur J. Jaworski, Acoustic Coupling Between the Loudspeaker and the Resonator in a Standing-Wave Thermoacoustic Device, Applied Acoustics, 2006, 402-419.

[22] A. C. Alcock, L. K. Tartibu, T. C. Jen, Experimental Investigation of Ceramic Substrates in Standing Wave Thermoacoustic Refrigerator, International Conference on Sustainable Materials Processing and Manufacturing (SMPM), Procedia Manufacturing,2017, 79-85.

[23] Bhansali P. S, Patunkar P. P, Gorade S. V, Adhav S. S, Botre S. S, An Overview of Stack Design for Thermoacoustic Refrigerator, International Journal of Research in Engineering and Technology (IJRET), 2015, 4, 68-72.

[24] B. AnandaRao, M. Prasanth Kumar, D. SrinivasaRao, Design and Experimental Study of Small-Scale Fabricated Thermo-Acoustic Refrigerator, International Journal of Engineering Trends and Technology (IJETT),2013, 4, 3830-3836.

[25] IkhsanSetiawan, AgungBambangSetio-Utomo, Makoto Nohtomi, Masafumi Katsuta, Experimental Study on Thermoacoustic Cooling System with Two Stacks in a Straight Resonator Tube, 10eme CongresFrancaisd'Acoustique, 2010.

[26] Kaushik S. Panara, AmratM.Patel. Jigar D. Patel, Thermoacoustic Refrigeration System Setup, International Journal of Mechanical Engineering and Technology (IJMET), 2015, 6, 01-15.

[27] Jithin George, Loud Speaker Driven Thermo Acoustic Refrigeration, International Journal of Scientific &Engineering Research, 2016,7, 465-468.

[28] Ramesh Nayak. B, Bheemsha, Pundarika. G, Performance Evaluation of Thermoacoustic Refrigerator Using Air as Working Medium, SSRG International Journal of Thermal Engineering (SSRG-IJTE), 2015, 1, 01-07.

[29] Hofler TJ. Thermoacoustic refrigerator design and performance. PhD Dissertation. Physics Department, University of California at San Diego; 1986.

[30] A. Widyaparaga, T. Hiromastu, T. Koshimizu, Deendarlianto, M. Kohno, Y. Takata, Thermoacoustic Heat Pumping Direction Alteration by Variation of Magnitude and Phase Difference of Opposing Sound Waves, Applied Thermal Engineering, Accepted Manuscript 2016.

[31] Yousif A. Abakr, Mushtak Al-Atabi, Chen Baiman, The Influence of Wave Patterns and Frequency on Thermo-Acoustic Cooling Effect, Journal of Engineering Science and Technology, 2011, 6, 392-396.

[32] HadiBabaei, Kamran Siddiqui, Design and Optimization of Thermoacoustic Devices, Energy Conversion and Management, 2008, 3585-3598.

[33] Florian Zink, Jeffrey S. Vipperman, Laura A. Schaefer, CFD Simulation of Thermoacoustic Cooling, International Journal of Heat and Mass transfer, 2010, 3940-3946.

[34] MatthieuGuedra, Flavio C. Bannwart, Guillaume Penelet, PierrickLotton, Parameter Estimation for the Characterization of Thermoacoustic Stacks and Regenerators, Applied Thermal Engineering, Accepted Manuscript-2015.

[35] MaximePerier-Muzet, Jean-Pierre Bedecarrats, Pascal Stouffs, Jean Castaing-Lasnignottes, Design and Dynamic Behaviour of a Cold Storage System Combined with a Solar Powered Thermoacoustic Refrigerator, Applied Thermal Engineering, Accepted Manuscript-2014.

[36] M. Nouh, O. Aldraihem, A. Baz, Piezo-Driven Thermoacoustic Refrigerators with Dynamic Magnifiers, Applied Acoustics, 2014, 86-99.

[37] J. A. Mumith, C. Makatsoris, T. G, Karayiannis, Design of Thermoacoustic Heat Engine for Low Temperature Waste Heat Recovery in Food Manufacturing – A Thermoacoustic Device for Heat Recovery, Applied Thermal Engineering, (2014), 588-596.

[38] Limin Zhang, Yanyan Chen, ErcangLuo, ANovel Thermoacoustic System for Natural Gas Liquefaction, The 6th International Conference on Applied Energy (ICAE), Energy Procedia, 2014, 1042-1046.

[39] Adeff JA, Hofler TJ. Design and construction of a solar powered, thermoacoustically driven thermoacoustic refrigerator. J Acoust Soc Am 2000; 107(6): L37-42.

[40] JIN, Tao, HUANG, Jiale, FENG, Ye, *et al.* Thermoacoustic prime movers and refrigerators: Thermally powered engines without moving components. *Energy*, 2015, vol. 93, p. 828-853.

[41] SreeneshValiyandi, Design and Fabrication of Thermoacoustic Refrigeration System, 4th International Conference on Science, Technology and Management (ICSTM-16), 2016, 236-245.

[42] N. Rott, “Thermoacoustics”, Advances in Applied Mechanics 20, 135 (1980).

[43] Li, Y. (2011). Thermoacoustic refrigerators : experiments and scaling analysis. Technische Universiteit Eindhoven. https://doi.org/10.6100/IR716451

[44] B. Higgins, Journal of Natural Philosophy and Chemical Arts 129 (1802), 22.

[45] A.A. Putnam and W.R. Dennis, Survey of organ-pipe oscillations in combustion systems, Journal of Acoustical Society of America 28 (1956), 246.

[46] P.L. Rijke, Notiz ¨uber eine neue Art, die in einer an beiden Enden offenen Rohre enthaltene Lift in Schwingungen zu versetzen, Annalen der Physik 107 (1859), 339.

[47] Lord Rayleigh, Theory of sound, Vol. II, Dover, New York, 1945.

[48] W.E. Gifford and R.C. Longsworth, Surface heat pumping, Advances in Cryogenic Engineering 1 (1966), 302.

[49] Thermoacoustic engines and refrigerators, Physics Today (1995), 22–28.

[50] J.C. Wheatley, T. Hofler, G.W. Swift, and A. Migliori, An intrinsically irreversible thermoacoustic heat engine, Journal of Acoustical Society of America 74 (1983), 153–170.

[51] Understanding some simple phenomena in thermoacoustics with applications to acoustical heat engines, American Journal of Physics 53 (1985), 147.

[52] T.J. Hofler, Thermoacoustic refrigerator design and performance, PhD Thesis, Physics department, University of California, San Diego, 1986.

[53] T.J. Hofler, J.C. Wheatley, G.W. Swift, and A. Migliori, Acoustic cooling engine, (1988), US Patent No. 4,722,201.

[54] P.H. Ceperley, A pistonless stirling engine - the traveling wave heat engine, Journal of Acoustical Society of America 66 (1979), 1508.

[55] Gain and efficiency of a short traveling wave heat engine, Journal of Acoustical Society of America 77 (1985), 1239–1244.

[55] H.B. Callen, Thermodynamics, ch. Appendix F, Wiley, New York, 1960.

[57] Panhuis, in 't, P. H. M. W. (2009). Mathematical aspects of thermoacoustics. Technische Universiteit Eindhoven. https://doi.org/10.6100/IR642908

[58] Swift, G.W., “Thermoacoustics: A Unifying Perspective for Some Engines and Refrigerators”, Acoustical Society of America, New York, (2002)

[59] Swift, G.W., “Thermoacoustic engines”, Journal of Acoustical Society of America 84, pp 1149-1152, (1988).

[60] Mr. Avishkar A Vadnere, Prof. Vasant Jog Final year students of Mechanical Engineering, Vice Principal G V Acharya Institute of Engineering & Technology, Shelu, Raigad- 410201, India

[61] F. Zink, J. S. Vipperman, L. A. Schaefer, “Environmental motivation to switch to thermoacoustic refrigeration”. Applied Thermal Engineering 30 (2010) 119-126.

[62] S. Humaira, T. Shohel, and R. A. Fraser, “Compressible pulsating convection through regular and random porous media : the thermoacoustic case,” pp. 329–342, 2012.

[63] I. Paek, J. E. Braun, and L. Mongeau, “Evaluation of standing-wave thermoacoustic cycles for cooling applications,” Int. J. Refrig., vol. 30, no. 6, pp. 1059–1071, 2007.

[64] M. E. H. Tijani, J. C. H. Zeegers, and a. T. a. M. de Waele, “Prandtl number and thermoacoustic refrigerators.,” J. Acoust. Soc. Am., vol. 112, no. 1, pp. 134–143, 2002.

[65] I. Paek, L. Mongeau, and J. E. Braun, “Performance characterization of a small-capacity thermoacoustic cooler for air-conditioning applications,” J. Mech. Sci. Technol., vol. 24, no. 9, pp. 1781–1791, 2010.

[66] NIRANTH M S, ELVIN LOBO, DEVRAJ P N, KIRAN LOBO. DESIGN AND fabrication of thermoacoustic refrigeration system. Visvesvaraya technological university Jnana Sangama, Belgaum-590014. 2013 – 2014.

[67] G Bharadwaj, G Dilip Kumar, G Teja, G Surya. Thermoacoustic Refrigeration. IJEDR 2019 | Volume 7, Issue 4 | ISSN: 2321-9939

[68] E. Tward, M. Petach, and S. Backhaus, Thermoacoustic space power converter, Space Technology and Applications International Forum, AIP Conference Proceedings **654** (2003), 656–661.

[69] Panhuis, in 't, P. H. M. W. (2009). Mathematical aspects of thermoacoustics. Technische Universiteit Eindhoven. https://doi.org/10.6100/IR642908

[70] CHINN, Daniel George. *Piezoelectrically-driven thermoacoustic refrigerator*. University of Maryland, College Park, 2010.

[71] M. A. Saad, *Thermodynamics: Principles and Practice*, Prentice-Hall, Inc. 95 (1997)

[72] DeltaEC users guide

[73] THERMODYNAMICS, Francis Wanjala Nyongesa, university of Nairobi, July 2015

[74] COUPER, James R., PENNEY, W. Roy, et JAMES, R. Fair, and Stanley M. Walas, 2012. Chemical Process Equipment Selection and Design, Third edition, Elsevier, united states of America.

[75] MORAN, Michael J., SHAPIRO, Howard N., BOETTNER, Daisie D., *et al. Fundamentals of engineering thermodynamics*. John Wiley & Sons, 2010.

[76] Hasan Aqeel *et al. Standing Waves Thermoacoustic Refrigeration System Using Loud-speaker as a Prime-Mover.* Senior Project Report University of Bahrain 10th of June 2020